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AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

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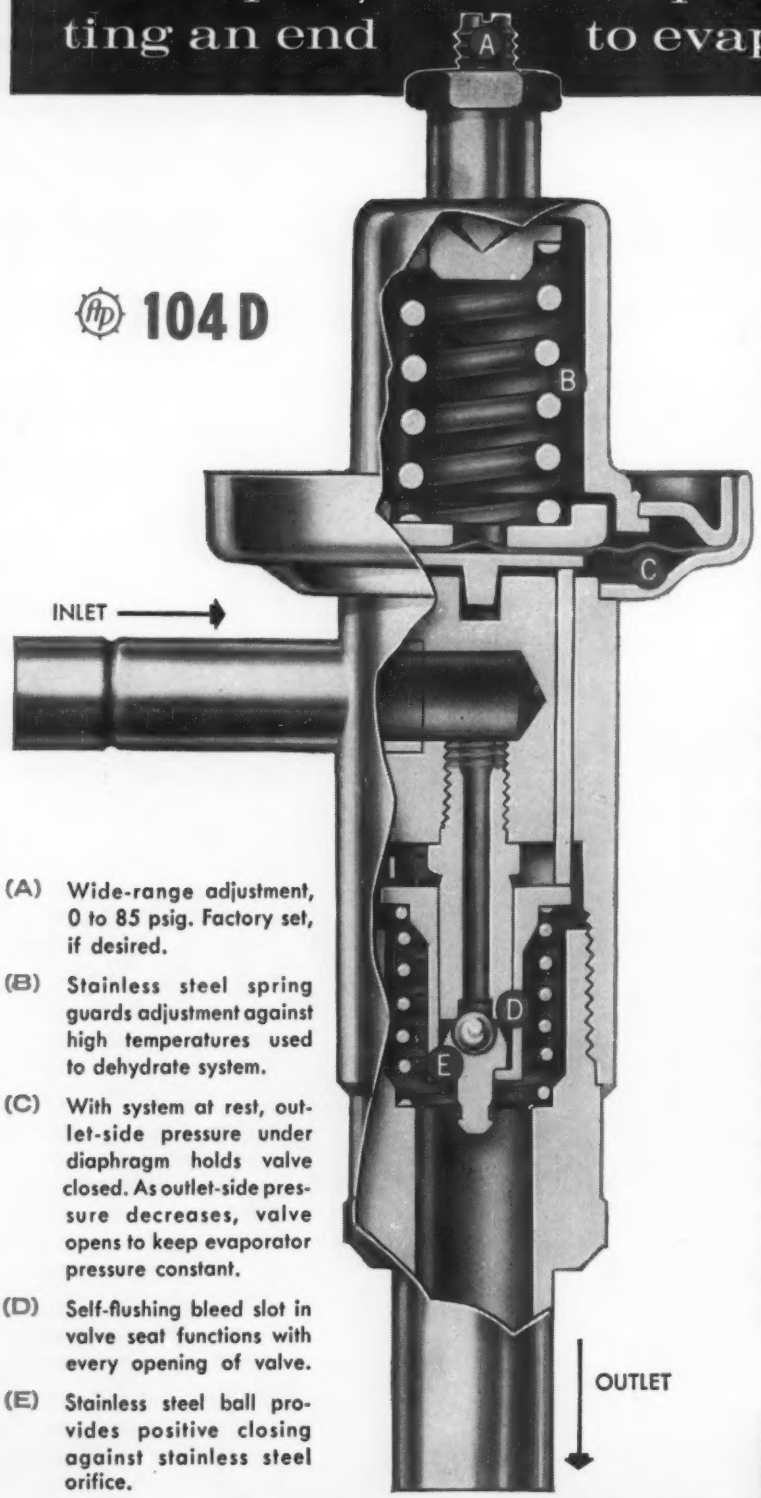
DECEMBER 1961

In room air conditioners and water coolers made by Carrier, Emerson, Fedders, Halsey Taylor, Westinghouse, Whirlpool and York . . . this new constant pressure valve protects their performance by putting an end to evaporator freeze-ups



104 D

(A) Wide-range adjustment, 0 to 85 psig. Factory set, if desired.

(B) Stainless steel spring guards adjustment against high temperatures used to dehydrate system.

(C) With system at rest, outlet-side pressure under diaphragm holds valve closed. As outlet-side pressure decreases, valve opens to keep evaporator pressure constant.

(D) Self-flushing bleed slot in valve seat functions with every opening of valve.

(E) Stainless steel ball provides positive closing against stainless steel orifice.

AND IT'S HERMETICALLY SEALED FOR FAST, EASY INSTALLATION

Here's a proven design for volume production — ready to go, nothing to assemble. And that's just the start of it, for this constant pressure valve eliminates evaporator freeze-ups regardless of load.

Triggers instant cooling action. The 104D stays closed during the off cycle . . . opens as required to keep evaporator pressure constant. With the valve bleeding off the high side pressure during off periods (within the two-minute UL specification), there's no chance for starting overloads, even with low-torque motors.

Ideal for other uses. The performance-protecting values offered by this new valve are not confined to room air conditioners and water coolers. With cost tuned to the needs of volume production, the 104D is ideally suited for use in cold beverage and food dispensers, ice cream cabinets, bottle and milk coolers, dehumidifiers. Whatever the installation, you can be sure of dependability that virtually eliminates service problems.

Controls Company of America is uniquely equipped to help you solve your control problems. A note from you will bring complete details about the AP valve (104D) featured here. Or ask for facts about other control opportunities.

Creative controls for industry

CONTROLS COMPANY OF AMERICA

HAC-14-60R

HEATING AND AIR CONDITIONING DIVISION

2456 N. 32nd Street, Milwaukee 10, Wis. • COOKSVILLE, Ontario • ZUG, Switzerland

DECEMBER
1961

VOL. 3 NO. 12

OFFICIAL PUBLICATION

JOURNAL

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Ross Gear and Tool Co., Lebanon, Tenn. Architects and Engineers: Spencer J. Warwick & Associates. General Contractor: Anderson and Gore. Mechanical Contractor: Central Air Conditioning & Heating, Inc.

680 tons of Chrysler Air Conditioning on the roof save 1100 sq. ft. of valuable floor space inside

Sprouting from the roof of this new plant for Ross Gear and Tool Company are 22 Chrysler 30-ton packaged air conditioning units. Together with two Chrysler packaged liquid chillers and four split-system units, they provide the cooling (785 tons of it!) for almost five acres of manufacturing and office space.

By putting 680 tons of Chrysler Air Conditioning *on* the roof, instead of *under* it, the company freed 1100 sq. ft. of valuable floor space for other jobs. And by installing these units zone-by-zone during construction, the company saved money by starting operations in part of the building while the rest was being completed.

The cooling load is spread over 28 different Chrysler units, ranging from a 4-ton condensing unit to a 45-ton chiller. Thus, varying departmental load requirements can be handled with ease. There is no fixed operating cost, as there would be with a conventional single-unit installation.

Whether your job calls for air-cooled or water-cooled equipment . . . packaged units or chillers . . . you'll find the complete Chrysler line has the exact unit you need. For complete data, or the technical cooperation of a Chrysler Engineer, write today.

CHRYSLER
AIRTEMP
A Division of Chrysler Corporation
Dept. V-121, Dayton 4, Ohio

Low Water Cut-offs on Hot Water Boilers?

...the industry has given the answer

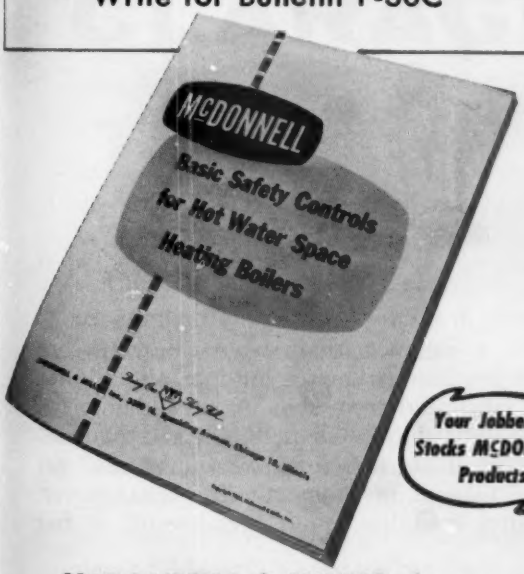
Yes, the answer has come from the field . . . from the engineers, contractors and manufacturers. They have seen the logic of installing a water level control on hot water space heating boilers . . . a low water cut-off, or—for even greater precaution—a feeder cut-off combination.

We knew it was a good idea. Expected it to grow and grow. But frankly we have been surprised at how rapidly the heating industry has taken hold of water level control as a logical team mate for an ASME pressure relief valve.

In fact, many local codes now require a low water cut-off, or feeder cut-off combination, on hot water boilers installed in places of public occupancy—including multiple dwelling units.

Notice the diagrams of recommended installations opposite. For more detailed discussion get this booklet that tells the whole story: "Basic Safety Controls for Hot Water Space Heating Boilers."

The What-Why-How in eight interesting pages
Write for Bulletin P-30C



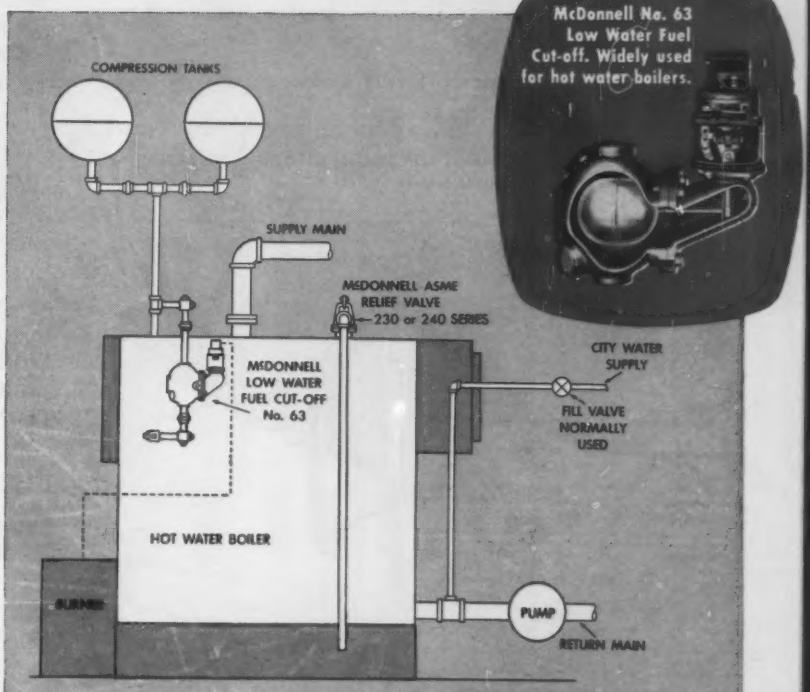
Your Jobber Stocks McDONNELL Products

McDONNELL & MILLER, Inc.
3500 N. Spaulding Ave., Chicago 18, Ill.

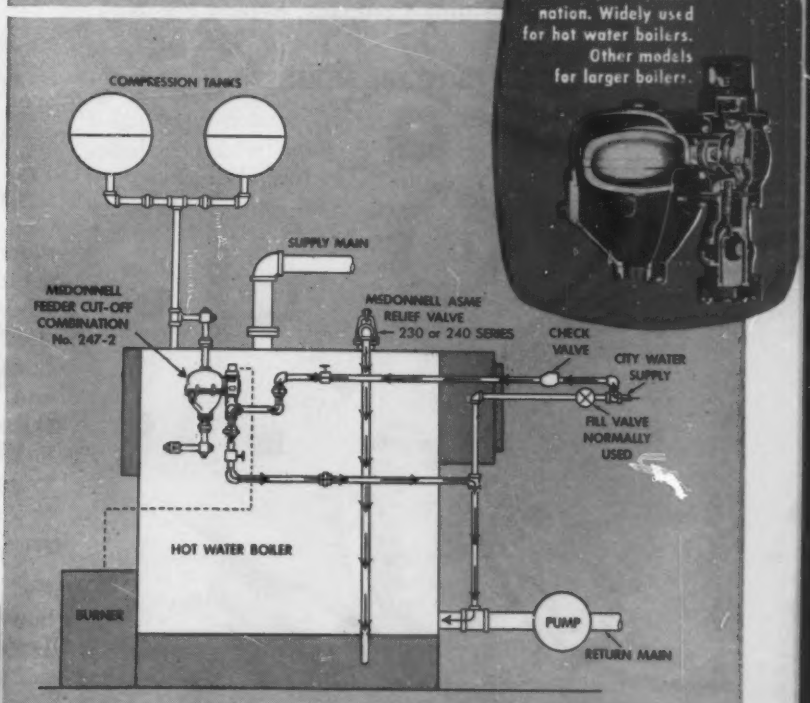
Doing One Thing Well

McDONNELL

Boiler Water Level Controls


McDonnell No. 63 Low Water Fuel Cut-off. Widely used for hot water boilers.
COMPRESSION TANKS
SUPPLY MAIN
McDONNELL ASME RELIEF VALVE 230 or 240 SERIES
McDONNELL LOW WATER FUEL CUT-OFF No. 63
CITY WATER SUPPLY
FILL VALVE NORMALLY USED
HOT WATER BOILER
BURNER
PUMP
RETURN MAIN

1 The essential low water control for a hot water boiler. A McDonnell Low Water Cut-off located above lowest permissable water level; also McDonnell A.S.M.E. Pressure Relief Valve.


McDonnell No. 247-2 Feeder Cut-off Combination. Widely used for hot water boilers. Other models for larger boilers.
COMPRESSION TANKS
SUPPLY MAIN
McDONNELL FEEDER CUT-OFF COMBINATION No. 247-2
McDONNELL ASME RELIEF VALVE 230 or 240 SERIES
CHECK VALVE
CITY WATER SUPPLY
FILL VALVE NORMALLY USED
HOT WATER BOILER
BURNER
PUMP
RETURN MAIN

2 The safest and most complete control for a hot water boiler. Offers a combination of mechanical and electrical safeguards which is today's best and most complete answer. A McDonnell combination Boiler Water Feeder and Low Water Cut-off; also McDonnell A.S.M.E. Pressure Relief Valve.

PARTS AND PRODUCTS

TEN-TON CONDENSING UNIT

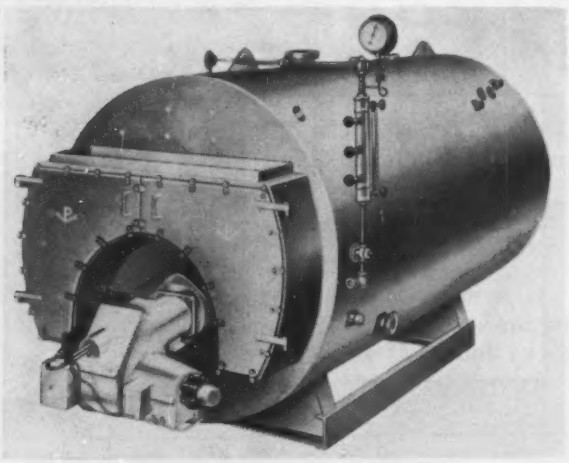
Capacity control is standard on this newly introduced unit, designed for rooftop installation. Weighing but

870 lb, the unit stands 20⅞ in. high without legs, which adjust to 12 or 18-in. heights. Vertical discharge propeller-type fans make possible the low silhouette and eliminate need for wind deflectors. Controls automatically unload two cylinders to give two-thirds capacity, delivering efficient cooling and humidity control under light loads. The Time Guard control circuit prevents rapid cycling of the compressor and reduces starting load on the electric system.

Carrier Air Conditioning Company, Syracuse 1, N. Y.

FORCED DRAFT BOILER SERIES

Offered is a wet-back constructed, Scotch-type boiler in a compact, forced draft package which includes boiler, burner and all controls. In forced draft design, minimum stack height is required, since the boiler



provides its own draft. Fired with No. 2 oil, natural, mixed or propane LP gas or combination oil-gas, the units are fully assembled, wired and factory-tested. Available in seven sizes, from 60 to 165 hp, boilers have a range of 2,000,000 to 5,540,000 Btu/hr and are low pressure: 15 psi steam, 30 psi water. To supply year round hot water, instantaneous or tank-type coils are available in a wide capacity range.

Portmar Boiler Company, Inc., 196 Seventh St., Brooklyn 15, N. Y.

RESIDENTIAL HUMIDIFIER

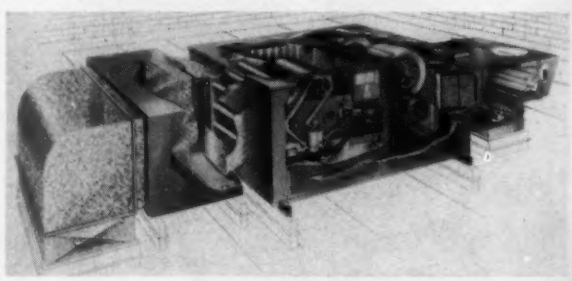
Designed specifically for small, warm air heated homes, the WF-J may be used with furnaces up to 100,000 Btu/hr. Operating on the Cold Steam principle and incorporating features of other units in the line, the humidifier provides positive evaporation of up to six gal/day. Motor operates on 35 watt, 110/120-volt ac.

Walton Laboratories, Inc., 1186 Grove St., Irvington 11, N. J.

FLEXIBLE ROOF-TOP UNITS

Offered with a choice of gas-fired, oil-fired or combination fuel models, units in this line are designed for the following types of applications: heating only, with or without filters; heating and ventilating, with or without filters, and with manual or automatic dampers to proportion fresh and return air; and heating-cooling-ventilating with manual or automatic dampers. Heating output range is 280,000 to 1,000,000 Btu/hr with an air volume range of 3000 to 15,000 cfm. Additive direct expansion evaporator units give 10 to 45 ton of cooling.

Combustion air and induced draft blowers are



mounted on a common shaft whose bearings are cooled by a continuous flow of outdoor air. Blower wheels are fabricated from stainless and aluminized steel and are welded to the shaft. Single and dual-fuel burners are available for No. 2 oil and natural gas and feature manual or automatic fuel changeover. Burner control includes on-off, high-low-off or full modulation.

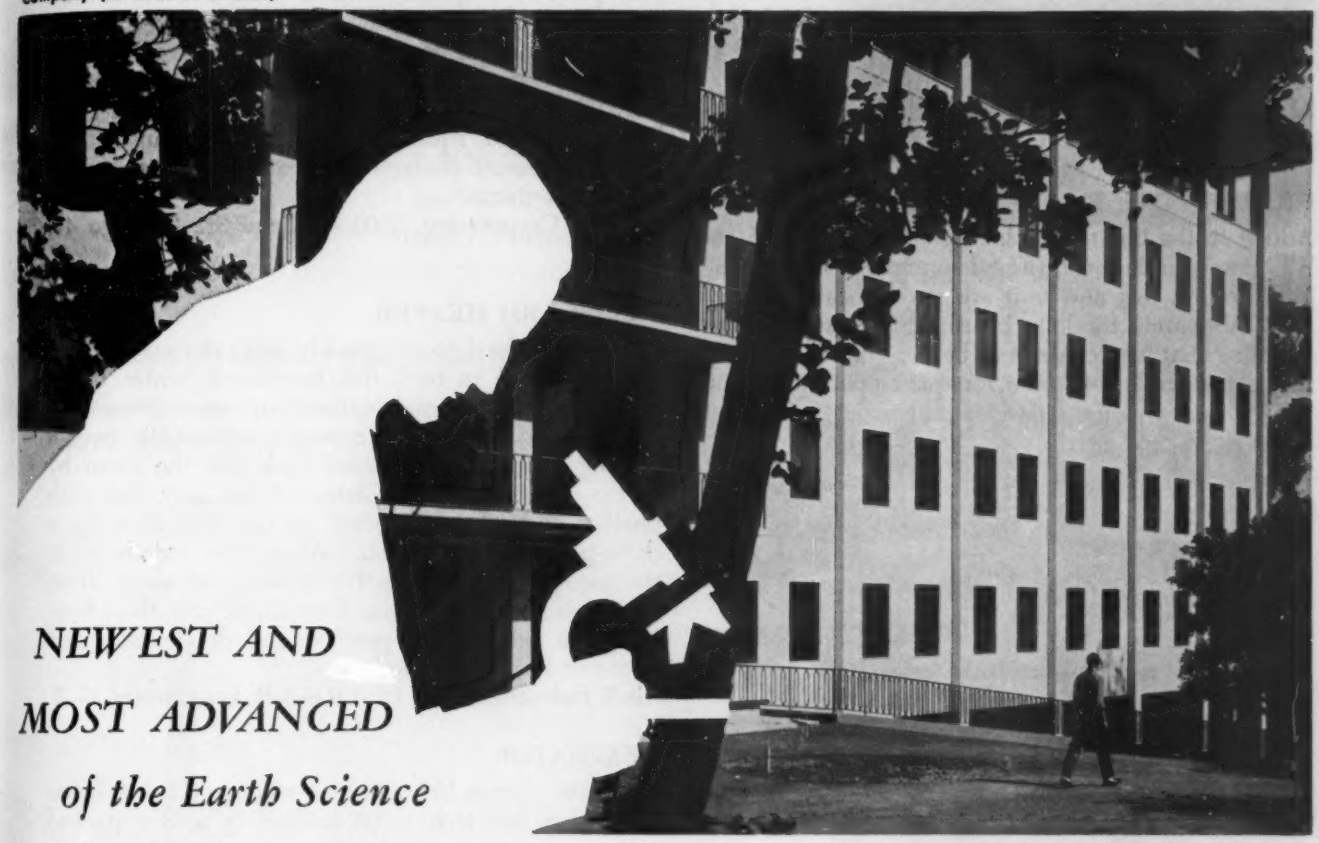
Lennox Industries, Inc., Marshalltown, Iowa.

HUMIDITY, TEMPERATURE INDICATORS

For measurement of temperature and humidity, two new flush-mounted instruments have been introduced, Durotherm Hygrometer Model No. AB2T and Thermometer Model No. TAB3T. Elements of the instruments project through a cabinet or room wall to a depth of one in. and measure the humidity or temperature of the area on the other side of the wall.

Applicable in temperatures up to 230 F, the hy-

ARCHITECTS: Warnecke and Warnecke. GENERAL CONTRACTORS: Rothschild, Raffin and Weirick, Inc. MECHANICAL CONTRACTORS: Herman Lawson Company. (All of San Francisco)



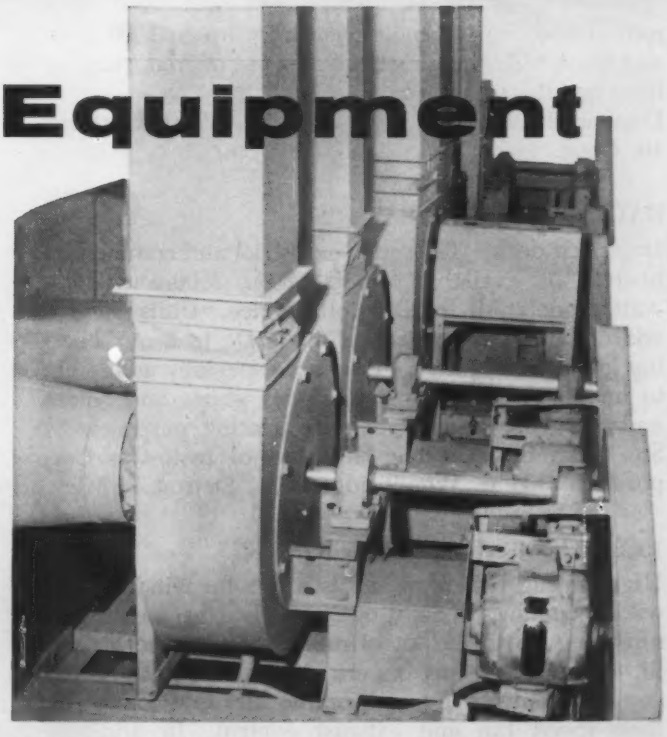
NEWEST AND MOST ADVANCED of the Earth Science Laboratories . . . uses

CLARAGE Equipment

ASSIGNMENT: AIR HANDLING in this new six-story, reinforced concrete structure for the Study of Geography, Geology, and Paleontology at a leading West Coast university.

Aiding students, staff, and classroom experiments are 39 Clarage Fans in general comfort and specialized laboratory hood exhaust applications.

Yes, for notable buildings the world over, Clarage air handling and conditioning equipment is a frequent choice because of its long-standing reputation for quiet, economical, dependable service. You too can profit by investigating the *quality* line we manufacture for buildings of all types, all sizes.



Largest of the 39 Clarage Fans installed on this particular job has a wheel diameter of 54¼" in double width construction. Shown here are a few of the smaller units in service.

Dependable equipment for making air your servant

CLARAGE FAN COMPANY

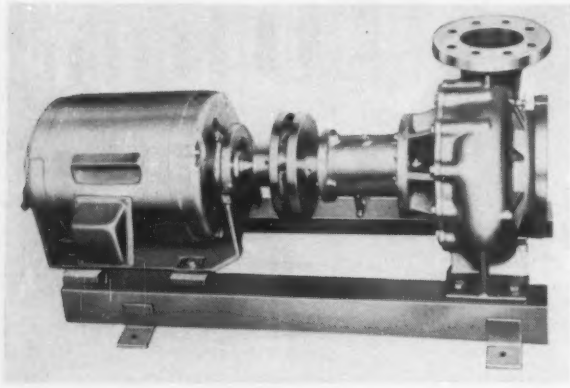
Kalamazoo, Michigan

SALES ENGINEERING OFFICES IN ALL PRINCIPAL CITIES • IN CANADA: Canada Fans, Ltd., 4285 Richelieu St., Montreal

grometer measures relative humidity from 0 to 100%. Dial diam is 5⅛ in. and casing is of solid brass. Range of the thermometer is from –30 to 130 F. Dial diam and casing are the same as for the hygrometer. Both are direct-reading.
Abrax Instrument Corporation, 179-15 Jamaica Ave., Jamaica 32, N. Y.

CENTRIFUGAL PUMP

Added to the line of In-Line Circulators and A5 and A9 close-coupled centrifugal pumps is Model B9. Featured on this new unit are sleeve bearing forced feed lubrication for long bearing life, a self-aligning coupling that connects sleeve bearing motor to pump, fabricated angle iron base for quiet operation, 1750-



rpm sleeve bearing motor, capacity up to 1000 gpm and heads to 80 ft. Eleven models are offered, ranging from one through ten hp.
Dunham-Bush, Inc., 179 South St., West Hartford 10, Conn.

RADIANT TUBE HEATERS

Improved design TD Series industrial and commercial heaters are available in 1500, 2000, 2500 and 3000-watt models, all for 240-volt service. Units have a wide range of use wherever moderate to heavy heating loads are required. They are especially adaptable to heating work areas in factories, shops and offices and for general supplementary heating purposes.
Sun-Heat, Inc., Electric Heat Div of Insto-Gas Corporation, 998 E. Woodbridge Ave., Detroit 7, Mich.

1962 ROOM AIR CONDITIONERS

Highlighting this new line is a 2½-hp window unit with a cooling capacity of 28,000 Btu/hr, capable of removing up to 7½ pt of moisture from the air per hr. Featured on this Imperial model are push-button comfort controls, five adjustable air flow louvers, a three-speed fan and exhaust control. In addition, there is a two-hp model rated at 23,000 Btu/hr.

Less than one ft deep, one-hp Ruler models can be installed rapidly with a Flex-O-Mount attachment. Range of their cooling capacity is from 5800 to 7200 Btu/hr, suitable for an average size room. Two models draw but 7½ amp; the larger unit with two-speed fan motor draws 10 amp.

More powerful, the one-hp Coronet series is rated at 9000 and 10,000 Btu/hr. Squirrel-cage fans have two speed adjustments. Chassis slides out and can be stored for the winter.

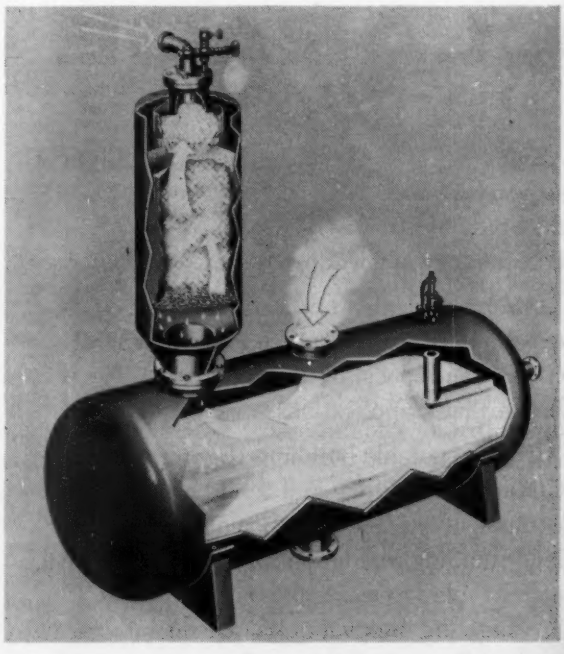
Two 1½ and one 2-hp air conditioners comprise the Royal series, operating on 230 volt. Units are rated at 12,000, 15,000 and 17,500 Btu/hr, respectively. They have two-speed, squirrel-cage evaporator fans, three-position exhaust controls and can be installed flush inside or outside.
Admiral Corporation, 3800 Cortland St., Chicago 47, Illinois.

BASEBOARD HEATER

Designed to heat rooms quickly and evenly at a cited saving of 20% in fuel, this baseboard heater has a built-in thermostat and features an ozone generating air purifier. Room air is moved mechanically over a heating coil and then forced back into the room by means of a Fan Base. Choice of fan only, fan with purifier and off is provided on the Fan Base by a three-position foot-switch. When the switch is in the second position, the fan draws cool air in from the room through a glass fiber filter, into the ozone generator where it is purified and deodorized, and over the heating element.
Air-X Industries, Inc., 1210 Randall Ave., Bronx, N. Y.

DEAERATOR

Corrosive oxygen in boiler feed water is cited as being reduced to less than 0.005 cc/liter by a new packed column deaerator. Water is spread into thin layers by means of a self-cleaning spray nozzle and a tray-



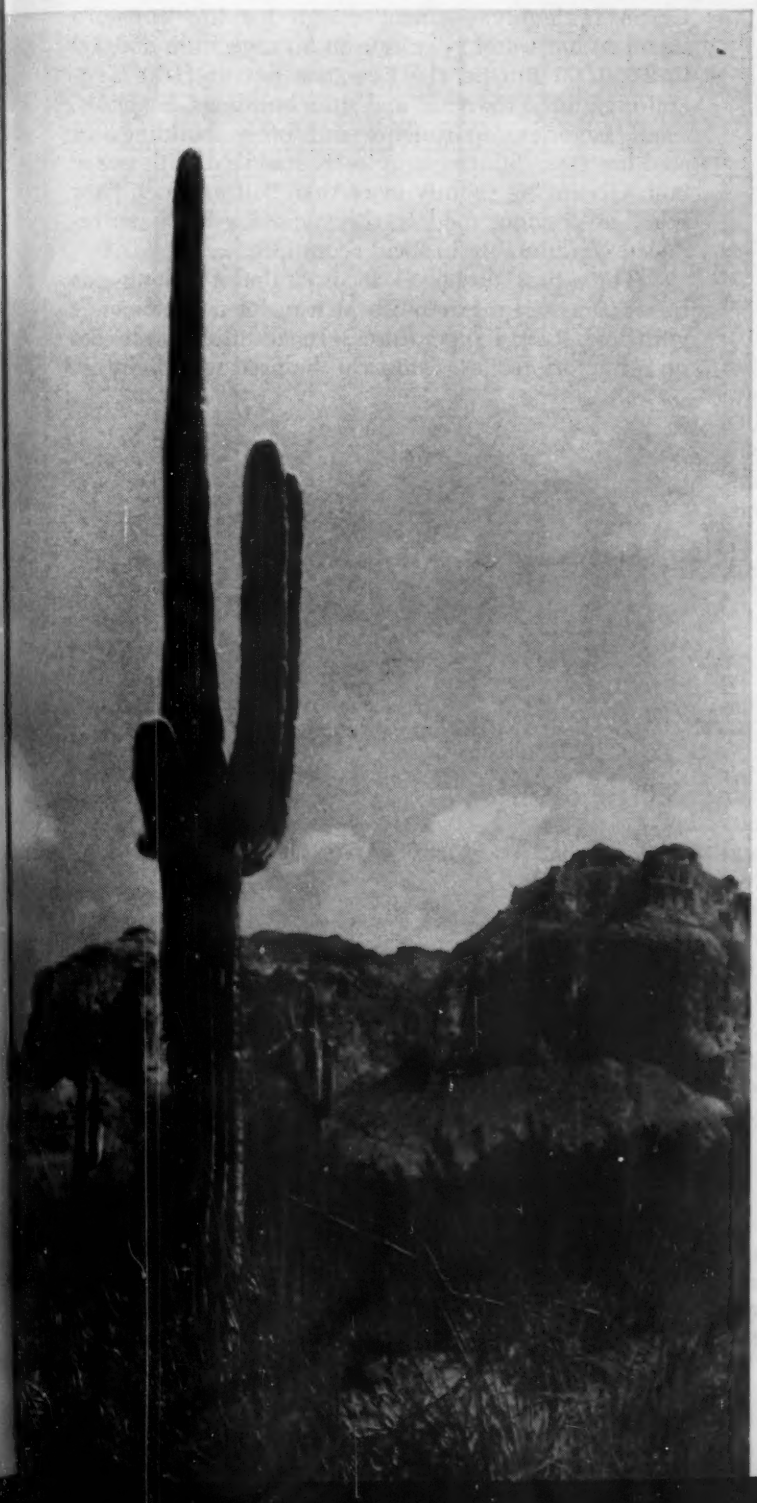
type exchange packing of great depth and surface exposure. Steam (white arrows in the illustration), moving counterflow to the water, scrubs out dissolved oxygen and carbon dioxide.

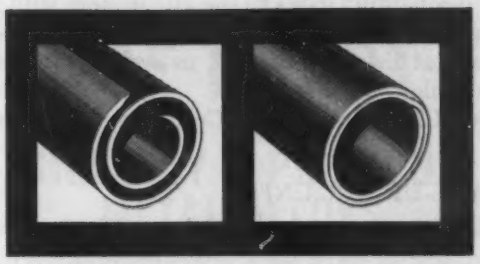
Seven models are available in a range from 1500 lb/hr at 50 F to 55,000 lb/hr at 150 F. Carbon steel storage tanks for deaerated water range in capacity from 53 to 1012 gal. An automatic variable vent is used

A case in point: refrigeration tubing. You name it, straight lengths, a simple bend or the most complex shape, Bundy has the facilities to precision mass-fabricate it at an attractively low unit cost. How does Bundy do it? . . . with tubing know-how plus all the advantages of working with Bundyweld® steel tubing. Bundyweld meets Government Specification MIL-T-3520, Type III; and ASTM 254. Standard wall thickness and O.D. are held to +.002″ to −.003″. It pays to come to Bundy for your refrigeration tubing requirements. Call, write, or wire: Bundy Tubing Company, Detroit 14, Michigan.

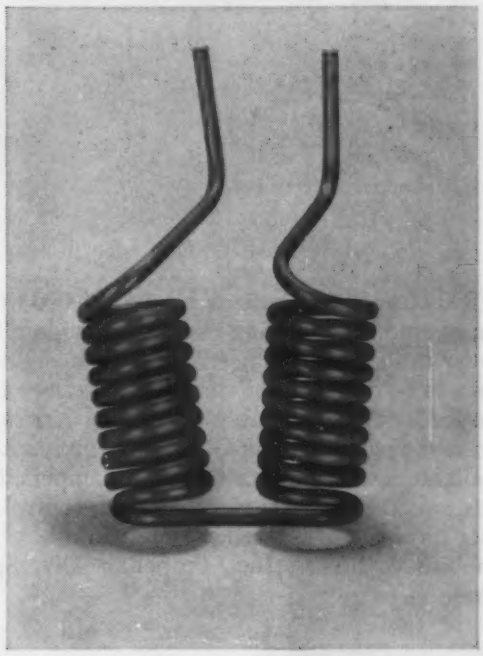
BUNDY TUBING COMPANY • DETROIT 14, MICH. • WINCHESTER, KY. • HOMETOWN, PA.

World's largest producer of refrigeration tubing. Affiliated plants in Australia, Brazil, England, France, Germany, Italy, Japan.





Bundyweld, double-walled from a single copper-plated steel strip, is metallurgically bonded through 360° of wall contact. It is lightweight and easily fabricated . . . has remarkably high bursting and fatigue strengths. Sizes available up to ⅝″ O.D.



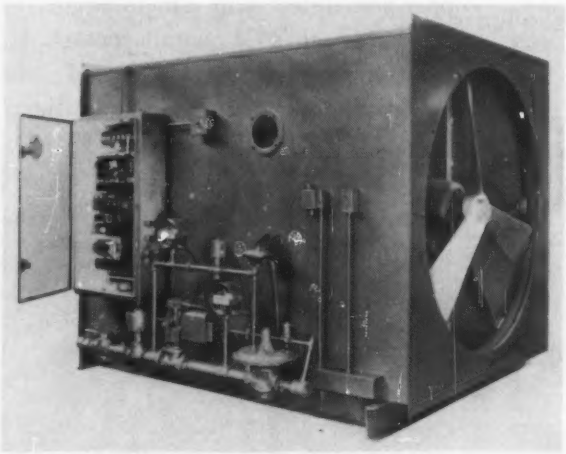
Ease of fabrication and high resistance to vibration fatigue make Bundyweld the choice for countless refrigeration applications. Seven feet of 3/16″ Bundyweld steel tubing is used in this oil cooler coil for a high-speed compressor. Notice the uniformity of the one-inch coils, possible because of Bundyweld's high quality.

BUNDYWELD® TUBING

for discharging gases to the atmosphere with minimum steam loss. Deaerator will operate on waste exhaust steam at five-psi pressure, as well as on steam from the boiler.
Aqua-Chem, Inc., 225 N. Grand Ave., Waukesha, Wisc.

MAKE-UP AIR UNITS

Self-contained and gas-fired, models in this line are available in three basic sizes, providing a total of six different ratings. Air deliveries range from 22,000 to



70,000 cfm; Btu/hr ratings from 2,125,000 to 5,375,000 are offered. Modulating controls provide a 25:1 turndown ratio and Summer-Winter controls enable the units to be used for hot weather breeze cooling. Combustion system is combined with an airfoil propeller fan into a compact, unit-housed assembly. Accessories available include: motor-operated dampers, filter cabinets, weather hoods, weatherproof control cabinets, duct turning elbows and remote control.
Robbins & Myers, Inc., Propellair Div, Springfield, Ohio.

1962 REFRIGERATOR LINE

Featuring models ranging from 9 to 15 cu ft, this line is highlighted by the Duplex, which has a full-size refrigerator and freezer side-by-side in one cabinet. Thin-wall construction with high density glass fiber insulation is featured on the entire line, and all units have full width freezer compartments with doors and porcelain crisper pans. Dual-Temp refrigerator-freezers are offered in right-side-up models with freezer compartment at the top and upside-down models with freezers at the bottom. There are three versions of the Duplex model: one never requires defrosting in the fresh food storage area, one has a self-defrosting freezer and another remains completely free of frost and features an automatic ice cube maker.
Admiral Corporation, 3800 Cortland St., Chicago 47, Illinois.

BALL VALVES

Available in sizes from ¼ through 2 in. for pressures to 1000 psig and temperatures to 550 F, ball valves in this line now are available with pneumatic and electric operators for automatic and remote on-off flow control applications. Both types of operators are sold integrally with the valve or separately for valves already in service.

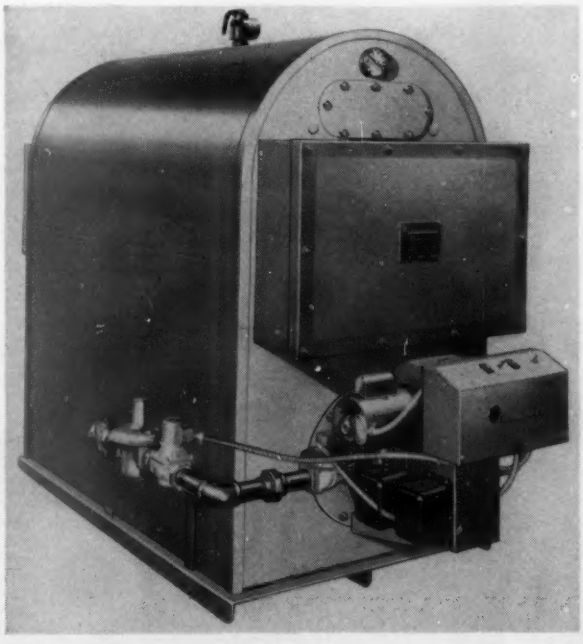
Pneumatic operators are available in three sizes and mount directly on the valve to simplify alignment and reduce space requirements. Optional accessories include a solenoid valve for electrical control and a limit switch for on-off indication.

Electric operators are of weatherproof and explosion-proof construction, featuring stainless steel output shaft, aluminum housing and hardened steel gear train. Units are clamp-mounted directly to the valve body. Cycle times as short as five sec from full open to full close are possible, depending on valve size.
Hills-McCanna Company, 400 Maple Ave., Carpentersville, Ohio.

COMPACT STEAM GENERATOR

Featuring highly compact design for low pressure steam or hot water operation in a range from 450,000 to 2,650,000 Btu/hr, the Kewanee Square-Heat Generator is suited for office and store buildings, hospitals, small factories, apartments and other buildings of medium size. Starting service is standard with every unit. Occupying slightly more than half as much floor space as previous models, the generator has been reduced considerably in head room also.

Three-pass design is incorporated for long gas travel to assure maximum heat transfer and efficiency with low stack temperatures. In addition, there are no refractory baffles. Units can be fired with light oil,



gas or a combination of both. Forced draft operation provides ample combustion air for stub stack applications. Special assemblies are available for high altitude applications.
American-Standard, Industrial Div, Detroit 32, Mich.

REMOTE TEMPERATURE CONTROLLERS

Two remote temperature controllers for use in air or liquid media, introduced recently, operate basically

BRAZING ALLOY REFERENCE CHART

SILVALOY BRAZING ALLOYS

SILVALOY	NOMINAL COMPOSITIONS					Solidus °F	Liquidus °F	Density troy oz /cu in
	Ag	Cu	Zn	Cd	Other			
5	5	88.75	0	0	P=6.25	1190	1300	4.37
15	15	80	0	0	P=5	1190	1300	4.45
35	35	26	21	18		1125	1295	4.90
45	45	15	16	24		1125	1145	4.92
50	50	15.5	16.5	18		1160	1175	5.00
503	50	15.5	15.5	16	Ni=3	1170	1270	5.00
355	56	22	17	0	Sn=5	1145	1205	5.00
I-401	5	0	16.6	78.4		480	600	4.53
377	5	0	0	95		640	740	4.60
A-4	9	53	38	0		1410	1565	4.50
20	20	45	30	5		1140	1500	4.64
A-11	20	45	35	0		1315	1500	4.68
A-79	25	52.5	22.5	0		1250	1575	4.71
A-13	30	38	32	0		1370	1410	4.67
250	40	30	28	0	Ni=2	1220	1435	4.76
254	40	30	25	0	Ni=5	1220	1580	4.72
A-14	40	36	24	0		1235	1415	4.80
105	45	30	12	0	Mn=13	1298	1298	4.76
A-18	45	30	25	0		1225	1370	4.82
A-25	50	34	16	0		1250	1425	4.99
54	54	40	5	0	Ni=1	1340	1575	5.06
A-33	60	25	15	0		1245	1325	5.02
60	60	30	0	0	Sn=10	1115	1325	5.23
T-50	62.5	32.5	0	0	Ni=5	1435	1590	5.19
Easy	65	20	15	0		1240	1325	5.06
Medium	70	20	10	0		1275	1360	5.14
301	72	28	0	0		1435	1435	5.24
AE-102	72	27.8	0	0	Li=.2	1410	1410	5.22
K-427	75	0	25	0		1300	1345	5.06
T-51	75	24.5	0	0	Ni=.5	1435	1475	5.30
Hard	75	22	3	0		1365	1450	5.28
T-52	77	21	0	0	Ni=2	1435	1525	5.32
A-49	80	16	4	0		1345	1490	5.29
AE-100	92.8	7.0	0	0	Li=0.2	1410	1610	5.46
S-475	95	0	0	0	Al=5	1550	1600	5.73
AT-601	97	0	0	0	Li=3	1115	1235	3.54
AT-600	98	0	0	0	Li=2	1290	1400	4.03

SILVALOY & PRECIOUS METAL ALLOYS

The American Platinum & Silver Division is ready to supply your brazing needs. The alloys listed here are part of the AP&S service. Refer your brazing problems to us for our recommendations, or write to American Platinum & Silver Division directly, or to your local Silvaloy distributor.

Silvaloy compositions include all commercially important silver base alloys and many special purpose compositions. The popular low temperature Silvaloy 35, Silvaloy 45, Silvaloy 15 are included in this group. If other compositions are required they can be supplied on special request.

We Recommend #1200 or Black Flux with Silvaloy Brazing Alloys

Ask for "A Complete Guide to Successful Silver Brazing...."

PRECIOUS METAL BRAZING ALLOYS

ENGALOY PM	NOMINAL COMPOSITIONS					Solidus °F	Liquidus °F	Density troy oz /cu in
	Au	Pd	Ag	Cu	Other			
478		5	68	27		1480	1490	5.31
491		10	58	32		1520	1565	5.30
490		15	65	20		1565	1650	5.45
493		20	52	28		1615	1650	5.41
492		25	54	21		1650	1740	5.51
428		5	95			1780	1850	5.56
431		10	90			1835	1950	5.60
447		20	80			1960	2150	5.67
485		30	70			2120	2250	5.74
238	80			20		1630	1630	8.28
261	75		5	20		1625	1640	7.99
265	72				(Ni=22) (Cr=6)	1785	1835	7.48
242	37.5			62.5		1814	1859	5.90
260	35			65		1832	1870	5.80
243	35			62	Ni=3	1787	1886	5.81
255	82				Ni=18	1742	1742	8.41
129		60			Ni=40	2260	2260	5.58
440		20	75		Mn=5	1830	2050	5.56
441		33	64		Mn=3	2100	2190	5.70

Precious metal compositions provide higher brazing temperatures as may be required for vacuum tube, aircraft, or missile use. These alloys will provide good corrosion resistance and minimum diffusion into base metals. They are ideal for atmosphere brazing.

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the same way: pressure on a remote liquid-filled sensing bulb changes with temperature to operate a snap-action switch. Units are available with one or two switches, the former for heating or cooling applications, the latter to offer sequencing over a temperature differential. Both models are available with fixed and adjustable temperature differentials. Three temperature ranges are offered: 0 to 100, 80 to 180 and 160 to 260 F. Units include an external adjustment knob with an easy-to-read scale, integral compensating mechanism for case ambient temperatures, single bulb size for all models and removable covers for easy access to wiring terminals.

Minneapolis-Honeywell Regulator Company, Commercial Div, 2753 Fourth Ave. S., Minneapolis 8, Minn.

1962 WINDOW AIR CONDITIONERS

Five lines are offered for 1962, including a new series of seven low-priced units, the Cavalier. Through-the-wall and standard window mounting kits are available for all models, and expandable wing panels are offered for all units up to 14 in. in depth. Two 115-volt Cavalier models offer 7000 Btu/hr on 7½ amp or 8500 Btu/hr on 12 amp; 208 and 230-volt units, in the same size casings, produce 9000 Btu/hr. Others in the series are a 230-volt model with a capacity of 9600 Btu/hr and two units with voltage of 208 or 230, available in 24-in. deep chassis and producing 14,000 Btu/hr. Exhausts and vents have been eliminated and air deflection is two-way instead of four, to reduce cost.

Sized for bedrooms, the 115-volt, 6000 Btu/hr Ready-Mount has reduced sound level and single on-off and thermostatic control located on the front. Units in the Sceptre line are available in 115-volt (with a choice of 7000 Btu/hr on 7½ amp or 9000 Btu/hr on 12 amp) and 208 and 230-volt, producing 9600 Btu/hr.

Offered in the Viscount series are 115-volt models with capacities of 8500 Btu/hr on 7½ amp and 11,000 Btu/hr on 12 amp; 208 and 230-volt models provide either 13,000 or 15,000 Btu/hr. Some units are available with a new hermetic compressor, the Micromite.

Illustrated here, the Imperial has a centrifugal evaporator fan cited as handling 470 cfm of air. Four-

row condenser coil permits an output of 20,000 Btu/hr with power consumption of but 2900 watt.

Heat pumps include a Sceptre series in 115 volt without supplemental strip heat and in 230 volt with an additional 5500 Btu/hr of electric strip heat capacity. A Viscount series, 230-volt unit gives 14,000 Btu/hr of cooling and adds strip heat of 6200 Btu/hr on heating. Provided by an Imperial heat pump are 20,000 Btu/hr of both cooling and heating. All models have automatic defrost.

Carrier Air Conditioning Company, Syracuse 1, N. Y.

AIR DIFFUSION SYSTEM

Developed for commercial, industrial and institutional buildings, this system utilizes a new type of ventilating acoustical ceiling which allows conditioned air to flow directly into a room at equal pressure and velocity through thousands of tiny perforations in the ceiling material itself. Since the entire ventilating ceiling distributes air, there are no drafts or stagnant spots.

Advantages over conventional air inlet systems

include elimination of conventional diffusers and a substantial amount of ductwork. In many cases, a single stub duct is all that is required to deliver the necessary volume of air to the plenum space above the ceiling. Conditioned air disperses through this space, builds up a slight pressure and is pushed down into the room through the perforations. Carbon dioxide gas is used in the illustration to demonstrate how air forms small jets beneath the ceiling, then diffuses in an overall blanket.

Ventilating ceilings are available in five different types of acoustical products: Travertone, Minatone, Minaboard, acoustical fire guard tile and acoustical fire guard lay-in units. Each offers sound absorption and incombustibility together with the air diffusion feature.

Armstrong Cork Company, Lancaster, Pa.

SWIMMING POOL HEATERS

Selection of pool water heaters on the basis of a three to four-day initial warm-up period is cited as adequately maintaining the pool at the desired tempera-

American-Standard Controls Division's new 900 expansion valve is compact, yet has the capacity to a big job. Part of the thermostatic power element is a length of copper capillary tube made by Wolverine Tube, Division of Calumet & Hecla, Inc. The tube helps form the sensing device that modulates the valve.

Wolverine small diameter control tube is selected because of its inherent quality and adherence to specifications. It is used not only by American-Standard Controls Division, but by many other control manufacturers.

Some prefer it in straight lengths, while others buy it in level wound coils that fit right into production lines. In the latter case, Wolverine can furnish control tube in lengths up to 4000 feet. It is available in sizes ranging from .062″ to .248″ O.D. with wall thicknesses from .010″ to .083″ maximum . . . and plain or electro-tinned.

Control tube is typical of the many products with which Wolverine has served the refrigeration and air conditioning industries for so many years.

Join today the list of leaders who depend upon Wolverine Tube. Just write—right now!

9722

WOLVERINE TUBE
DIVISION OF
Calumet & Hecla, Inc.
DEPT. T, 17244 SOUTHFIELD RD., ALLEN PARK, MICH.
TUBEMANSHIP in Copper—Copper Alloys—Aluminum—Special Metals

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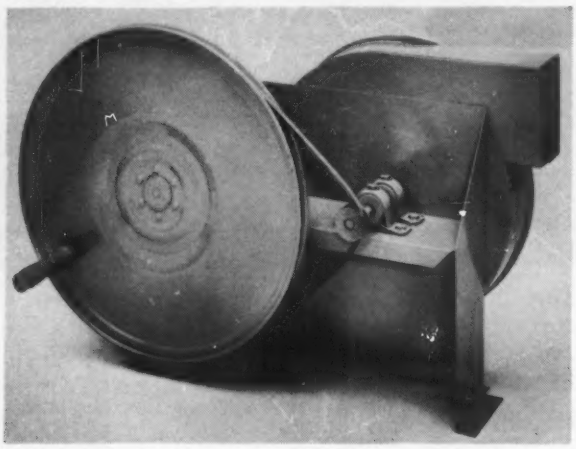
ture when air temperature is 20 to 25 F below the water temperature, if the filter pump is operated from 16 to 24 hr a day. Based on these qualifications, the Copper Sanimaster Model CS 18-100 will heat a pool with 400 sq ft of surface area or approximately 16,000 gal of water; Model CS 37-150 will heat a pool with 700 sq ft of surface area or approximately 29,000 gal of water.

Units are designed specifically for use as swimming pool heaters and provide a compact installation without the need for a large storage tank. Control of the heater is by an adjustable thermostat which has a two-in. temperature differential and which actuates the water heater whenever pool temperature drops below the desired level. Construction of the tank is of copper, reinforced with steel.

Rheem Manufacturing Company, 7600 S. Kedzie Ave., Chicago 52, Ill.

FALL-OUT SHELTER VENTILATION

Hand-powered and cited as being mechanically foolproof, this fall-out shelter ventilation system comes



in a packaged unit that includes a filter. Ventilation capacity is enough for shelters housing ten to twelve persons. Motor-driven models also are available.

Omaha Air Products, P. O. Box 68, West Dodge Station, Omaha, Nebr.

ROOF-TOP UNITS

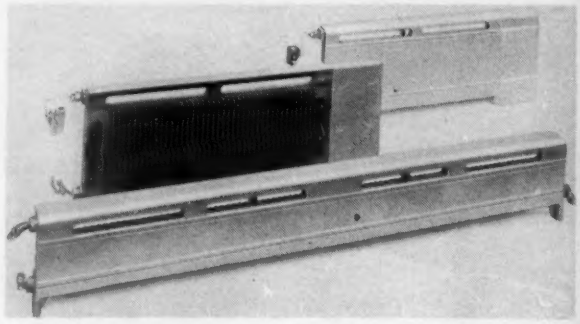
Combining heating and cooling, the WAGH Series was designed and engineered for roof-top installation. All units are pre-piped and wired at the factory and delivered with a refrigerant test charge. For maximum strength, the equipment is mounted on rigid iron channels. Foil-backed, glass fiber insulation is used throughout. Permanent-type, expanded aluminum filters are standard. The electrical system includes a 24-volt control panel with a master disconnect circuit breaker.

Constructed of heavy gauge aluminized steel, the furnace section includes an automatic draft diverter, built-in stack switch, automatic pressure regulator and gas burner with auto-ignition pilot and main gas shutoff.

Welbilt Air Conditioning and Heating Corporation, Farmingdale, N. J.

HEATING SYSTEM

Basically, this new development is a hydronic heating system using cast baseboard radiation with electrical

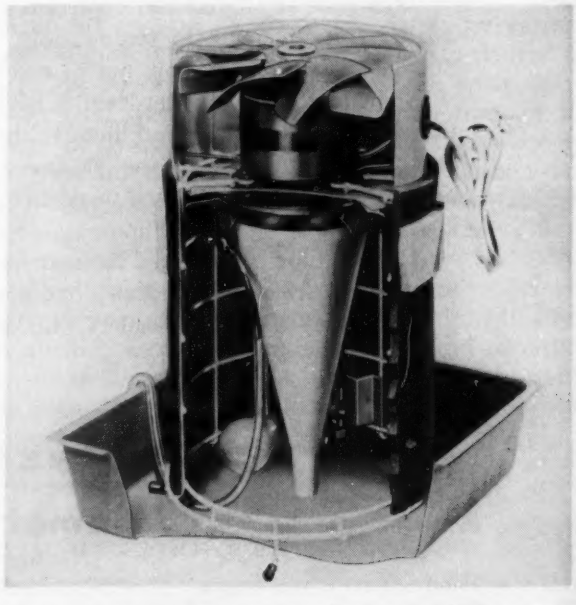


energy supplying the heating power. Electric heat producing element is within the radiation chamber itself, with the liquid in the chamber transferring the energy from this element to the radiant heating surface. Basic to the system is a central control system, cited as enabling control of temperatures to within ¼ F, room-by-room or zone-by-zone. A low-mass wall thermostat in each room or zone provides rapid, frequent heat signals to a heat anticipating relay in the master control panel. This thermostat-relay combination anticipates comfort requirements. Comprising the entire system are individual baseboard panels with separate heating elements, master control panel, a small expansion tank that fits between the floor joists and a reducing and relief valve combination. Shown are front and back views of an assembled H panel and front view of the L panel.

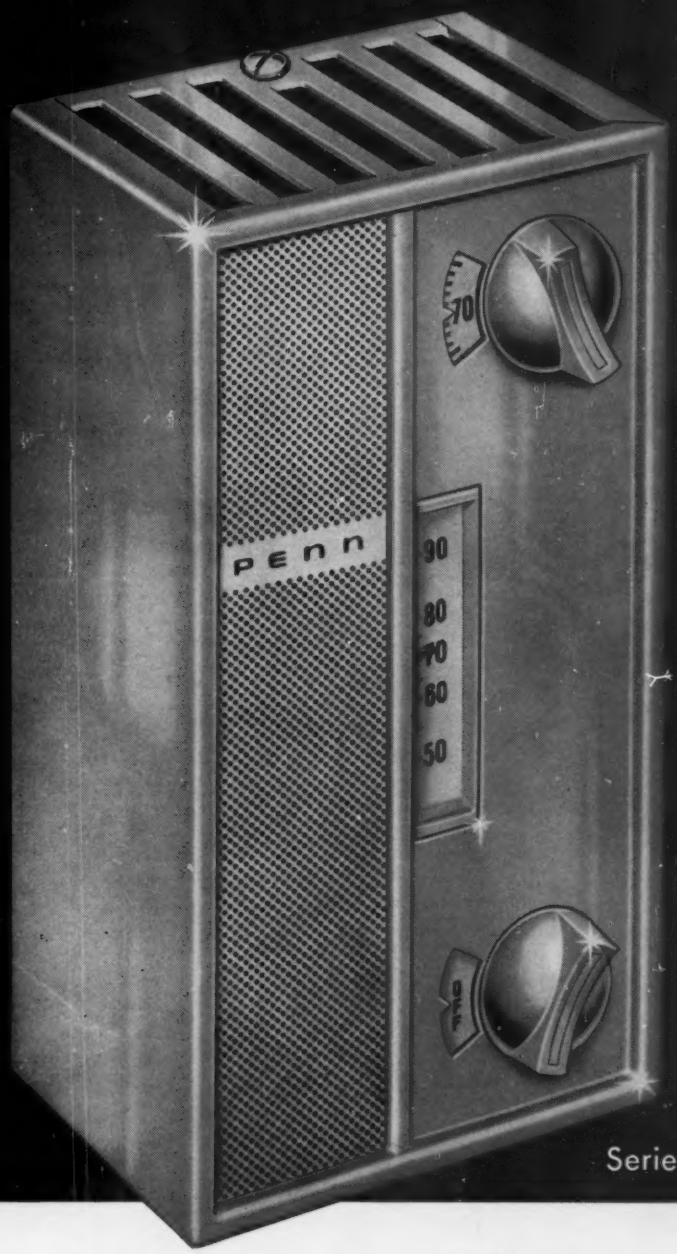
Aqua-Lectric, Inc., Minneapolis, Minn.

HUMIDIFIER

Evaporating a high volume of water, the Mist-I-Cone humidifier is cited as also trapping dust, water minerals and water soluble odors. Centrifugal action of a spinning cone picks up water from a reservoir pan



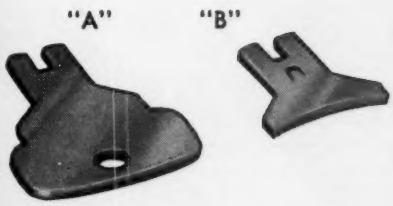
and throws it against a breaker comb. This water saturates a high porosity evaporating mat enclosing



NEW

line voltage THERMOSTAT with front panel control of all heating-cooling functions!

Series 822



How settings are locked

Where it is desirable to lock settings, installer removes cover and pushes out factory-installed key "B" from each adjusting knob. This disconnects knob from adjusting shaft. Then, user may insert special key "A" (supplied for key chain use) through knob slot and turn to make desired settings . . . remove key and thermostat is locked as adjusted.

Take your choice...heating, cooling or heating-cooling...each features unique, easy-to-use, locking-type knobs

Here's the line voltage thermostat that has "everything" desired for accurately controlling temperature in motels, schools, offices and similar installations.

It has easy-to-read temperature and function selector dials. It has an accurate bimetallic thermometer with vertical dial. It has exclusive, locking type adjusting knobs, attached to cover but connected to adjusting shafts by factory-installed "keys" within the knobs. It has a unique, built-in but concealed adjustable high limit stop for temperature dial. It has an unusually sensitive element and mechanism which provide low operating differential at all voltages and all amperages within its rating. It has easiest "field wiring" ever...simply connect to large terminals *on back of thermostat*. And, it has modern "thin-line", compact styling. Learn more about this new thermostat...write to the Penn factory for Bulletin 3233.

PENN CONTROLS, INC. Goshen, Indiana

EXPORT DIVISION: 27 E. 38th ST., NEW YORK, N.Y.

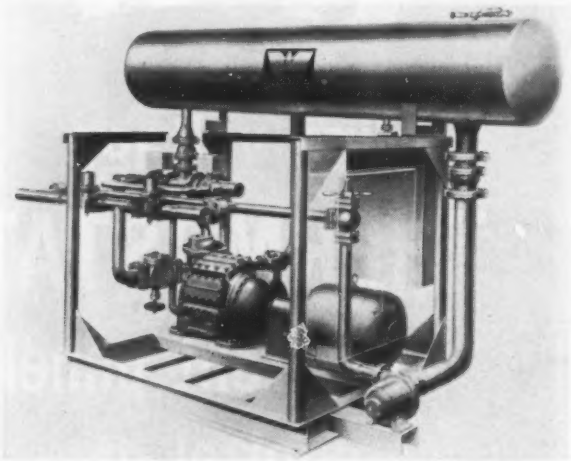
AUTOMATIC CONTROLS FOR HEATING, REFRIGERATION, AIR CONDITIONING, APPLIANCES, PUMPS, AIR COMPRESSORS, ENGINES

the unit. A self-contained fan draws dry air into the unit's interior and forces it out through the saturated mat. Air leaving the mat is cited as containing 90 to 95% humidity.

Associated Mechanical Services, Inc., 688 Jenks Ave., St. Paul, Minn.

COMMERCIAL HEAT PUMPS

Manufactured specifically for use in the 20 to 300-ton field, these units are based on positive forced feed



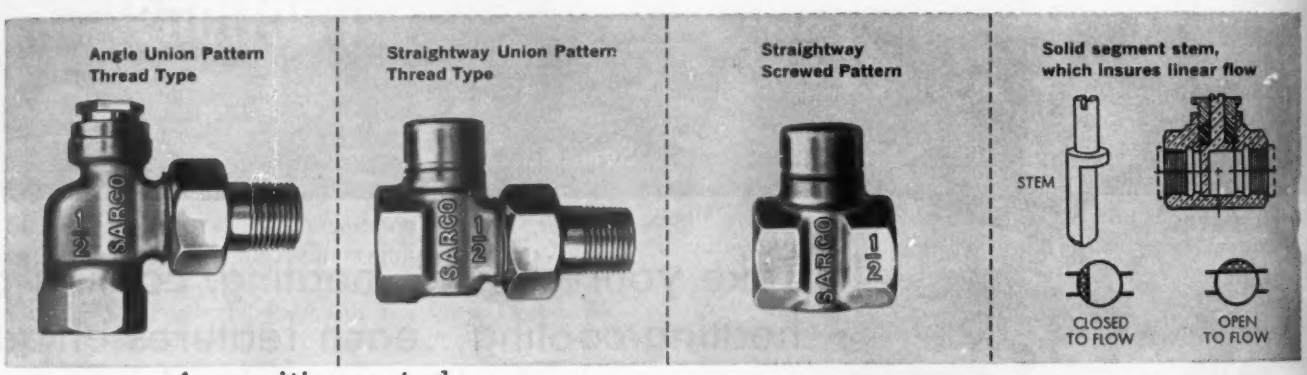
circulation of refrigerant through the summer or winter evaporator. No critical superheat control settings govern its operation. Compressors are protected from slugging during operation, switching and defrosting. Eliminated in the design are three and four-way valves, expansion valves, oil separators and oil receivers and many refrigerant controls. Units are available for both air-to-air (shown) and air-to-water operation, replacing conventional heating applications.

Worthington Corporation, Air Conditioning Div, East Orange, N. J.

RESIDENTIAL AIR CONDITIONERS

Pre-assembled and charged with refrigerant at the factory, Model 1254 consists of an evaporator coil, 24 ft of refrigerant tubing and a condensing unit. Installation work has been reduced to mounting the evaporator on the furnace, drilling a three-in. hole through an outside wall, uncoiling the refrigerant tubing from the evaporator case and twisting the coupling on the end of the tubing to a coupling on the outside condensing unit. Capacity of the unit is 35,000 Btu/hr and it is capable of air conditioning up to 2000 sq ft.

Another whole-house air conditioner introduced recently is the Model 1140, a 33,000-Btu/hr unit capable of serving up to 2500 sq ft. Consisting of the evaporator and condensing coils and the compressor, the unit circulates air through the conditioned space by means of the furnace fan. Main installation requirements are supply and return air ducts between the unit and the furnace.



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in balancing hot water systems, use **SARCO**

SOLID SEGMENT STEM BALANCING FITTINGS

Sarco Balancing Fittings use a stem having a segment of a circle, instead of the butterfly stem on old type balancing fittings. Sarco construction gives you three solid advantages:

First, you get linear flow. Equal openings give you even throttling action and that can save you hours in balancing water systems. Second, when you set a Sarco fitting with this solid segment stem in the full open position you get no obstruction to flow. That means less pressure drop through the fitting. Third, Sarco balancing fittings are designed to operate perfectly to 200 psi maximum – cold liquid, non-shock.

Available in both thread and sweat types

Sarco Balancing Fittings are available in both thread and sweat types, in all patterns. Also with integral air vents. Ready for prompt delivery, too: access boxes for balancing fittings; valves; air eliminators. Write for Bulletin. 7076

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News highlights of the month

ERRATUM

Attention of ASHRAE membership is called to an unfortunate error on page 72 of the October issue of the JOURNAL under the section With The Committees, "To Discontinue Chapter Allocations (June 1962)."

Quoted herewith is the exact wording of the motion passed by the Board of Directors at its meeting on June 29, 1961:

7. VOTED: THAT CHAPTER ALLOCATIONS BE DISCONTINUED AS OF JUNE 1962; WITH THE PROVISION THAT THE REGIONS CENTRAL COMMITTEE FIRST EXAMINE THIS THROUGH THE CHAPTERS AND REPORT BACK TO THIS BOARD IN JANUARY 1962, SUBJECT TO THE APPROVAL OF THE CHAPTERS.

TRENDS

New standard An industry certification program for room fan-coil air conditioners is scheduled to become operative in October 1962. The Air Conditioning and Refrigeration Institute, the Air Moving and Conditioning Association, Inc., and the Institute of Boiler and Radiator Manufacturers cooperated to develop a new standard, designated "Industry Standard 441," which provides the basis for the program. The program will be administered by ARI. The Fan-Coil Coordinating Committee, representing manufacturers of this type of equipment from the membership of the three sponsoring organizations, will act as an advisory group to consider matters of general policy within the program.

ASHRAE lectures November 16 began a series of six weekly lectures on "Sound and Vibration Control for Air Conditioning and Ventilating Systems" sponsored by the New York Chapter of ASHRAE. The lectures, held at the United Engineering Center, encompass such topics as noise control criteria, selection and specifications of materials, analysis of noise producers and vibration isolation and control. Speakers are Laymon Miller, Bolt Beranek & Newman, Lewis Goodfriend, L. Goodfriend & Associates and Michael Kodaras, Acoustical Consultant. Additional information about the series may be obtained from Alfred Greenberg, 90 Laurel Hill Terrace, New York 33, N. Y.

INDUSTRY CHANGES

JETS move JETS, a national junior engineering society for high school students, has moved from Michigan State University to the United Engineering Center, 345 East 47th Street, New York, N. Y. Having recently established affiliation with the professional engineering societies, JETS has almost 900 chapters throughout the country and over 20,000 members.

BRI change For ten years a part of the National Academy of Sciences-National Research Council, the Building Research Institute will become an independent technical society sometime next year. While continuing to function as a membership organization and to carry out existing policies and activities, it is contemplated that the change will enable BRI to broaden further its operations in the field of building research.

BOOK REVIEWS

Hevac handbook A 204-page handbook, containing the proceedings of the Heating, Ventilating and Air Conditioning International Conference held in London, September 27 to October 4, has been issued. The book includes brief summaries of the papers given, biographical notes on the authors, open discussion on conference papers, the conference committees and the daily programs. The conference was sponsored by the Institution of Heating and Ventilating Engineers, 49 Cadagon Square, London, S.W.1, in cooperation with the Association of Heating, Ventilating and Domestic Engineering Employees and the Heating and Ventilating Research Association.

Activities report Issued in a 34-page booklet is the activities report for 1960 of the U. S. National Committee for the International Institute of Refrigeration, National Academy of Sciences-National Research Council, Washington 25, D. C. Included in the report are the International Institute of Refrigeration Commission Meetings to be held in the U. S. in 1962, membership and representation of IIR Technical Commissions, committee activities and reports of Committee members who are officers of IIR.

Load calculation Offering a single load calculation procedure that has been industry-accepted is the aim of a new manual published by the National Warm Air Heating and Air Conditioning Association, 640 Engineers Building, Cleveland 14, Ohio. The publication, Manual J – "Load Calculation for Residential Winter and Summer Air Conditioning," has three separate divisions which include explanations of the load calculating procedure, tables and maps, and factors governing unusual construction features. $2.50.

Ionized air According to a recent Air Force study, breathing air with excesses of positive or negative ions has no discernible effect on human attitudes or ability to perform complex mental tasks. The study is reported upon in a 24-page booklet entitled "A Study of the Effects of Ionized Air on Behavior," available from the U. S. Department of Commerce (Order PB 171 600 from OTS), Washington 25, D. C. $.75.

ASTM Proceedings Now available from the American Society for Testing and Materials is its 1242-page ASTM "Proceedings," Volume 60, the official record of the transactions of the Society for 1960. The volume consists of 44 technical papers on subjects pertaining to research and standards for materials, summaries of several symposiums held by ASTM technical committees, lists of all symposiums and other special sessions published separately as Special Technical Publications and all papers published in the ASTM monthly bulletin. Copies of the book may be obtained from the American Society for Testing and Materials, 1916 Race Street, Philadelphia 3, Pa. $12.

SPECIAL MEETINGS

ASME 82nd Held November 26-30, New York, the 82nd Annual Meeting of the American Society of Mechanical Engineers consisted of a comprehensive program, including a jointly-sponsored symposium with the American Society of Heating, Refrigerating and Air Conditioning Engineers on the subject of psychrometrics. Highlight of the Psychrometrics Symposium was the introduction of a new psychrometric chart recently adopted by ASHRAE. Also of interest was a luncheon honoring the 50th anniversary of Dr. Willis H. Carrier's publication of "Rational Psychrometric Formulae."

Comfort heating Scheduled for March 19-21, 1962, in Chicago, is the second National Electric Comfort Heating Exposition and Symposium, sponsored by the National Electrical Manufacturers Association, 155 East 44th Street, New York, N. Y., which will include technical symposiums and an exhibition of the latest equipment, materials and developments in the electric comfort heating industry. Some of the products that will be displayed include heat pumps, insulation and insulation equipment, controls, radiant heaters and electric steam radiators.

ENGINEERING MAGIC...'DDEC'

DUAL-DRY Evaporative Condenser

DRY FAN DRY CASING

Here's the newest, most imaginative design concept the refrigeration industry's seen in years! The Dual-Dry Evaporative Condenser features blow-through dry fan construction; and the 'DDEC' is designed with a dry casing. The recirculating water is completely enclosed in a molded fiberglass liner and never contacts the outer steel casing.

Because the pump and water piping as well as the fans and fan motor are all factory installed within the dry casing the contractor saves installation time and money.

This fine unit incorporates many other superior features:

- Pump and water piping, fans, fan motor and drive are factory assembled within the casing. Thus, installation costs are minimized, protection provided for long service-free life.
- Coils are all prime surface copper. Eliminators and water piping are plastic.
- Dual casing and sound absorbing qualities of molded fiberglass provide unusually quiet operation.
- No steel in contact with the water in any part of the unit.

Capacities range from 10 to 100 tons.

Form 7020, free on request, contains complete specifications.

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DUNHAM-BUSH, INC.

WEST HARTFORD 10, CONNECTICUT, U. S. A.

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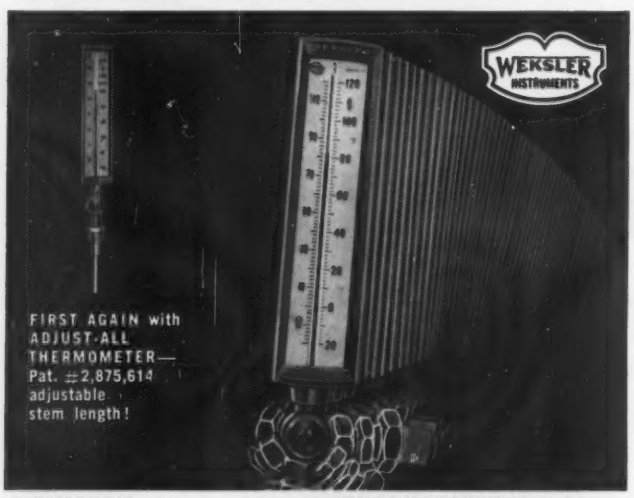


A MILLION *Adjust-Angle* THERMOMETERS AGO...

...takes us back thirteen years, to U. S. Patent No. 2,524,894 awarded to Weksler. Until a few months ago, Weksler was the only adjust-angle thermometer you could buy. Today you have a choice from several different manufacturers who are in first model production of this remarkable instrument.

Thirteen years' experience with a precise instrument represents a world of know-how...a decisive consideration in your choice of Weksler! Write for Bulletin AA-60.

New Weksler adjust-angle industrial thermometers, completely redesigned, are full 2-inch width...easy reading...highest accuracy in any range you desire. 7- 9- 12-inch scale sizes, plus exclusive 5" scale size!

WEKSLER INSTRUMENTS
"ORIGINATORS OF WORLD RENOWNED ADJUST-ANGLE THERMOMETERS"
WEKSLER INSTRUMENTS
CORPORATION
195 EAST MERRICK ROAD, FREEPORT, L. I., N. Y.
Indicating and Recording Instruments for Temperature, Pressure and Humidity

For southern climate conditions, a packaged heat pump, Model 3103, is designed to operate in summer temperatures of 120 F and winter ambient air of 10 F. Drain pan heaters extend operation to colder northern temperatures. Capacity is 26,000 Btu/hr for both heating and cooling. Application is to most types of installation: roof, attic, basement, carport or the ground. Flush mounting in an outside wall also is possible.

Chrysler Corporation, Airtemp Div, 1600 Webster St., Dayton 4, Ohio.

IMPINGED JET BURNERS

With capacities ranging from 400,000 to 3,200,000 Btu/hr, these conversion burners are available with

a choice of three control systems: standing for 24 volt, standing electronic and intermittent full electronic for 115 volt. Flame is distributed to the heat transfer surface of boilers or furnaces as a low level, intensely hot blanket. Burners can be turned down to as low as 0.1 in. wc pressure and still burn with a stable flame without flashing back.

Barber Manufacturing Company, 1052 E. 134th St., Cleveland 10, Ohio.

NYLON PRESSURE HOSE

Strong, flexible and with a low refrigerant effusion rate, Nylaflow refrigerant hose is cited as having lost but ½ gm/ft of Refrigerant 12 in the first seven days of testing with no subsequent losses and 9 gm/ft of Refrigerant 22 in seven days of testing. Permanently bonded and all-nylon, the hose consists of a polyamide innertube with an overbraid of high tensile strength cord and an abrasion-resistant external cover. Hose design permits a low bend radius and firm attachment of compact fittings.

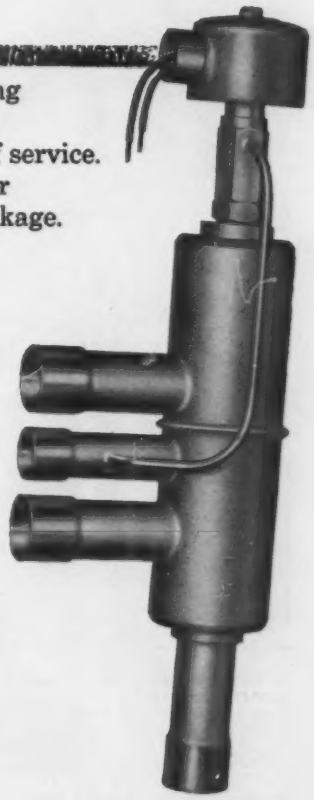
Rated at 4000 psi-minimum burst strength, the hose is available in IDs of 3/16 through ¾ in. and is supplied in continuous lengths from 150 to 500 ft or in assemblies with reusable couplings. In addition to its light weight and high strength, Nylaflow hose operates in temperatures to 225 F and is virtually

Buy SPORLAN

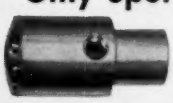
REVERSING VALVES
THERMOSTATIC EXPANSION VALVES
REFRIGERANT DISTRIBUTORS
CATCH-ALLS
SEE • ALLS

for HEAT PUMP *peak performance*

Sporlan Reversing Valves the reliable 4-way change-over control with the new Floating Ring design. Proven synthetic seating material assures complete tightness under all operating conditions throughout many years of service. No close fitting metal-to-metal sliding seals to wedge, corrode or wear, with resultant erratic performance or by-pass leakage.



Sporlan Thermostatic Expansion Valves are specifically designed for heat pump applications with elements made to meet their unusual operating requirements. Rugged in construction to withstand high pressures . . . with diaphragms designed for long life. Special selective charges maintain control of the system under extreme variations of ambient temperatures.



Only Sporlan Offers Refrigerant Distributors with auxiliary side connections for reverse cycle applications . . . plus the versatile interchangeable nozzle that gives you peak coil performance on heat pumps.

The Sporlan Catch-All is the perfect filter-drier for heat pumps. Ample water capacity...unexcelled in acid removal...with the famous molded porous core that retains those extra fine particles which cause abrasive damage or troublesome leakage. Only Sporlan Catch-Alls give you really clean, really dry, acid free refrigeration systems.

With a Sporlan See•All you can check the refrigerant in a heat pump system at a glance, because it's the moisture and liquid indicator with the single dot that shows how dry the refrigerant is... and the full view sight glass that lets you see if the system is fully charged.

Contact us today for additional engineering data

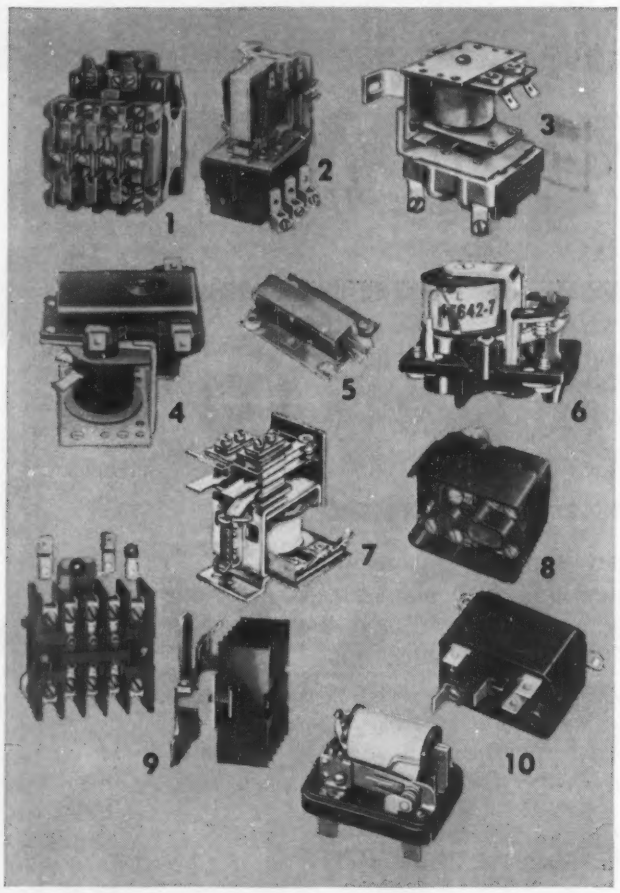
SPORLAN VALVE COMPANY
7525 SUSSEX AVENUE ST. LOUIS 17, MISSOURI

EXPORT DEPARTMENT, 85 BROAD STREET, NEW YORK 4, N. Y.

only RBM

...has engineered a complete control family for

AIR CONDITIONING



1. TYPE C-30, C-40, C-50 CONTACTORS
2-3-4 pole 30 amp.—to 600 volts; 2-3-4 pole 40 amp.—to 600 volts; 2 pole 50 amp.—230 volts. Same mounting holes and coils for all ratings and pole forms.
Write for Bulletin 1070.

2. TYPE S-30, S-40 CONTACTORS TO 600 VOLTS
2-3 pole rated 30 or 40 amps. Low cost, small size. Exceeds rigid requirements of industry's largest users.
Write for Bulletin C-10.

3. TYPE 80 CONTROLLER
Specific design for nominal 3 HP or 3-ton single phase compressors.
Write for Bulletin C-11.

4. TYPE 75 POWER RELAY
Low cost, dependable. Handles up to 6000 W. at 240 V., resistive load per pole. Compressor rating 2 poles, 18 amp. running, 90 amp. locked rotor at 250 volts.
Write for Bulletin 1030.

5. BIMETAL TIME DELAY RELAYS AND THERMOSTATIC DEVICES
To 230 volts. NO-NC. Common or isolated heater. Quick connect terminals.
Write for Bulletin C-15.

6. SERIES 129000 SHUNT TYPE RELAY — SPNO, SPNC OR NO-NC
For standard commercial voltages. Other coils available for special application. For heater, fan control, general circuit switching, etc.
Write for Bulletin 1010-A.

7. GENERAL PURPOSE RELAY
98000 Series AC or DC. Permits engineering short cuts lowering "finished product" cost.
Write for Bulletin 1060.

8. SERIES 128000 POTENTIAL STARTING RELAY
For starting single phase capacitor start compressors.
Write for Bulletin 1010-A.

9. TYPE 85, 25 AMP—2, 3 POLE CONTACTOR
Simplifies panel circuitry, helps eliminate terminal boards, cuts wiring costs.
Write for Bulletin C-16.

10. TYPE 83 AND 84 ENCLOSED BANTAM RELAYS
Series type motor starting and General Purpose single pole switching.
Write for Bulletin 1080.

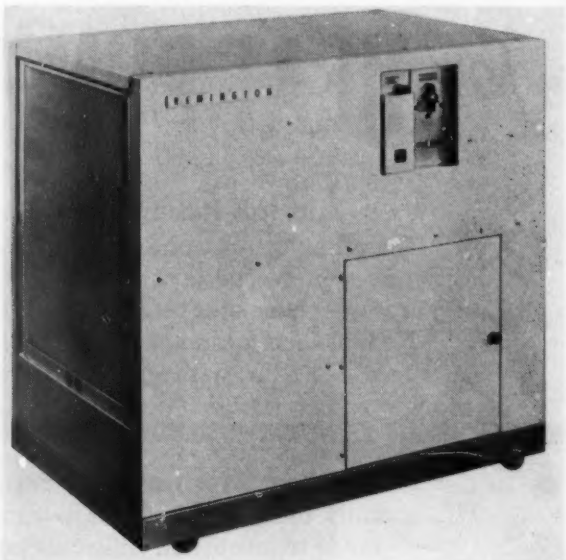
For the most efficient and low cost solution of your control problems, consult your local RBM Product Application Engineer.

RBM Controls Division
LOGANSPORT, INDIANA

unaffected by hydraulic fluids, oils, greases, solvents or alkalis.
Polymer Corporation, 2120 Fairmont Ave., Reading, Penna.

INDUSTRIAL DEHUMIDIFIER

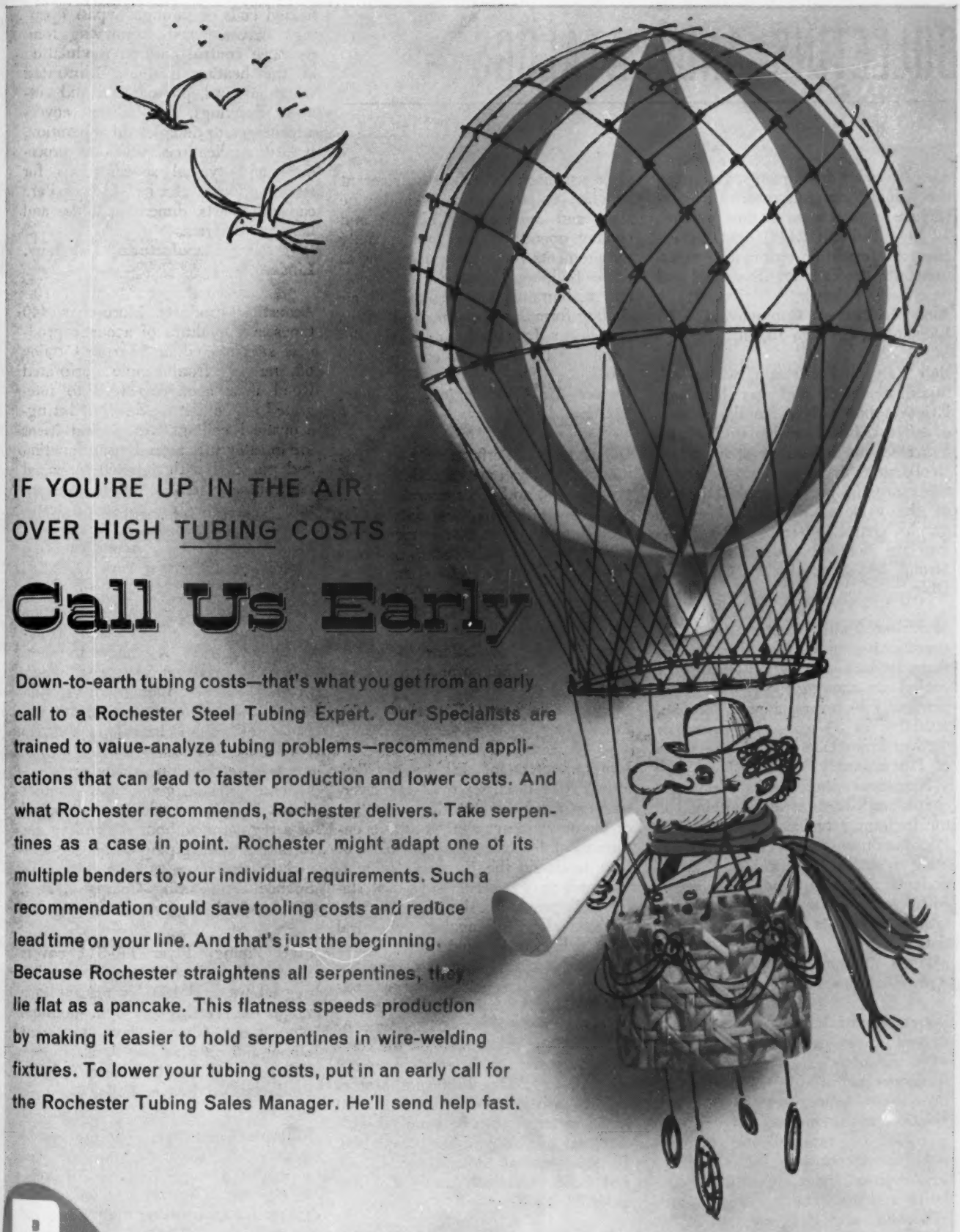
Operating on the same principle as household types in the line, heavy-duty Model D20 has more than ten times the moisture removing capacity, up to 300 lb daily. Utilized is a two-hp, hermetically sealed refrigeration cycle. Units are recommended for use in ambient temperatures from 50 to 110 F and relative humidities down to 40%. Standard accessories are: electric humidistat with magnetic starter, Freezestat for lower operating temperatures, four ball-bearing casters, a pail compartment complete with five-gal pail for locations where a drain is not available and a twelve-ft, three-wire electric cord. Models are available for 208 or 230-volt, 60-cycle, or 200 or 230-volt, 50-cycle, single-phase current.



Remington Corporation, Air Conditioning Div, Auburn, N. Y.

LOW TEMPERATURE CONTROLLER

Automatic control for cryogenic liquid control applications is provided by Model UP 1015, an ultrasonic level control instrument. Of stainless steel, the probe is activated when any cryogenic liquid touches its sensitive surface. The unit has repeatability within thousandths of an in. and is not affected by clinging droplets. Capable of operating at a pressure over 2000 psi, the probe does not actuate in foam or froth. No adjustment is required for temperature shifts or pressure variations. Ultrasonic probes can be used interchangeably in different fluids with varying dielectrics and conductivities. Application is for overflow alarm, propellant utilization systems, automatic tank filling applications, low level alarms and engine cut-off sensors.
Powertron Ultrasonics Corporation, Expressway Industrial Park, Fairchild Ave., Plainview, N. Y.


IF YOU'RE UP IN THE AIR
OVER HIGH TUBING COSTS
Call Us Early
Down-to-earth tubing costs—that's what you get from an early call to a Rochester Steel Tubing Expert. Our Specialists are trained to value-analyze tubing problems—recommend applications that can lead to faster production and lower costs. And what Rochester recommends, Rochester delivers. Take serpentines as a case in point. Rochester might adapt one of its multiple benders to your individual requirements. Such a recommendation could save tooling costs and reduce lead time on your line. And that's just the beginning. Because Rochester straightens all serpentines, they lie flat as a pancake. This flatness speeds production by making it easier to hold serpentines in wire-welding fixtures. To lower your tubing costs, put in an early call for the Rochester Tubing Sales Manager. He'll send help fast.
RP
STEEL TUBING BY ROCHESTER PRODUCTS
ROCHESTER PRODUCTS DIVISION OF GENERAL MOTORS, ROCHESTER, NEW YORK

BULLETINS AND CATALOGS

Air Measuring Instruments. Featuring the Velometer and accessories, a comprehensive release on air flow systems, 12-page Bulletin 2448, contains extensive information on air flow measurement. Products are illustrated and described in detail.
Alnor Instrument Company, 420 N. LaSalle St., Chicago 10, Ill.

Ball Valves. For steam, water, oil, liquid chemical and gas handling, Evertyte ball valves are the subject of a flyer. Made from bar stock (stainless steel or cadmium-plated carbon steel), units have an easy-to-service top entry design. An exploded view of the valve is shown and photographs, charts and engineering drawings are included.
Strong, 508 Sandusky St., Conneaut, Ohio.

Multizone Units. Central station air conditioning units, described in 64-page Bulletin 34, supply heated or cooled air simultaneously and independently to several zones. They are available in 34 sizes, horizontal or vertical arrangement and three types of fan assembly.

Numerous photographs, drawings, charts and diagrams are contained in the bulletin, which provides detailed information on construction and operating features, description of components, weights and dimensions, coil selection, fan performance, coil ratings, K factors and other technical data.
Marlo Coil Company, 7100 S. Grand Ave., St. Louis 11, Mo.

Boiler Feedwater Treatment. Answered in 24-page Bulletin 30 are such questions as: "How pure must feedwater be?", "How does operating pressure influence boiler water composition requirements?" and "What is meant by 'external' and 'internal' feedwater treatment?" Included is information on water impurities, coagulation and precipitation, ion exchange processes, and data used in water chemistry.
Nalco Chemical Company, 6216 W. 66th Pl., Chicago 38, Ill.

Reference Guide. Contained on both sides of a 3½ x 6½-in. plastics-enclosed card are key data used by designers of air conditioning systems, consulting engineers and architects. Included are a psychrometric chart, refrigerant weights, volume, enthalpy and pipe sizes and approximate heating and cooling operating costs and capacity requirements for such applications as office buildings, apartments, schools and restaurants. Conversion and air cooling formulas, sheet metal weights per linear ft and formulas for determining fixed charges complete the card.
Carrier Air Conditioning Company, Syracuse 1, N. Y.

Data-Loggers. Performing two basic functions – systematic collection of process, test and other operating data and presentation of these data in the most useful form for supervisory, process analysis and research purposes or for direct input into computers– Data-Master loggers are the subject of six-page Bulletin D401. Photographs supplement discussion of operating system installations in manufacturing, distribution and process industries.
Bristol Company, Waterbury 20, Conn.

Motor Protection. Eight-page Bulletin F2034 discusses solid-cast Everseal encapsulation for the protection of motor windings against severe environmental conditions. Results of tests made on the Everseal formulation and on completed solid-cast stators are provided, showing the protection afforded against chemical fumes, heat and cold, dust, water splash, salt spray, abrasives, repeated and prolonged electrical surges, solvents, acids, caustics and other environmental conditions. This type of encapsulation is available on the company's motors with random-wound coils, up to 125 hp, 1800 rpm. Motors with form-wound coils (150 hp, 1800 rpm and larger) can be supplied with Everseal silicone-rubber insulation.
U. S. Electrical Motors, Inc., P. O. Box 2058, Terminal Annex, Los Angeles 54, Calif.

Heater Sections. For air conditioning preheat, fresh air supply and process and drying applications, heater sections with a special face-and-bypass damper system for temperature control are described in 12-page Bulletin IFB-61. Discussed is how thermostatically-controlled dampers regulate the proportion of air passing over heated coils or through bypass openings between coils, achieving temperature control without modulation of the heating medium. Illustrated with diagrams, photographs and cutaway drawings, the bulletin covers advantages, principles of operation, typical applications, selection procedure and typical specifications for steam and hot water use. Included are capacity charts, dimension tables and hook-up diagrams.
L. J. Wing Manufacturing Company, Linden, N. J.

Acoustical Products. More than 440 types and qualities of acoustic products are covered in 30-page Catalog 62, ranging from simple perforated wood fiber acoustical tiles to integrated air conditioning-ventilating-acoustical ceilings. Individual items are in categories according to function and purpose, with detailed technical information and illustrations. A separate section deals with sound transmission loss problems.
Elof Hansson, Inc., Acoustical Div, 711 Third Ave., New York 17, N. Y.

Air-Control Regulators. Application engineering for efficient use of air-control pressure regulators is published in a series of guide sheets (A5B10). Hook-up diagrams, recommended installation and steam capacity tables are included. How to obtain efficient steam use for an absorption refrigeration machine is described in one guide and another shows how to provide remote control of a regulator without consuming air, rerouting main piping or adding special air piping.
Spence Engineering Company, Inc., Walden 2, N. Y.

Steel Piping. Flyer TF-511 covers short radius, double extra strong, 90-deg elbows and 180-deg returns in a range of 1½ through 8-in. nominal pipe size. Charts show pertinent elbow and return dimensions.
Babcock & Wilcox Company, Tubular Products Div, Beaver Falls, Pa.

Multiple Connectors. Three separate lines of multiple connectors are described in 12-page Bulletin 467-20M-PP-61, which provides specific data sheets for each of the lines. Provided are extensive specifications on contact sizes and materials, wire crimp and insulation support, wire and insulation size ranges, receptacles and housings.

Fastin-Faston multiple connectors, containing one, two, six or eight separate and fully insulated circuits, are used in both the appliance and auto-

there IS a way

to cut heating/cooling piping costs and still have a piping job you can be proud of

Yes, the new Taylor Forge Compact Line of welding fittings really does reduce piping costs. First, because the simplified designs and reduced dimensions permit major manufacturing economies. Second, because all other incidental costs—shipping, storage, trucking, on-the-job handling—are also less than with conventional fittings.

Take the 4" size for example: A "set" of Compact Fittings consisting of a 90° and 45° elbow, straight tee, cap, 4" x 2" tee, 4" x 2" concentric reducer and 4" x 2" eccentric reducer exactly as illustrated here, weighs little more than half as much as corresponding conventional fittings. And remember, this weight savings is purely the result of designing and dimensioning functionally for economy.

Taylor Forge Compact Line Fittings give you a piping job you can be proud of, too. Proud of its trim functional appearance. Proud of its rugged strength and complete dependability—well supported by the test described on the opposite page.

Compact Line Fittings are of seamless construction using Grade B carbon steel. The line includes 90° and 45° elbows, straight and reducing tees, concentric and eccentric reducers and caps in sizes 2", 2½", 3", 4", 5" and 6"; reducing tees and reducers in intermediate iron pipe sizes down to one-half of the nominal run or large end size. They are manufactured to the same high standards as other Taylor Forge products for piping and pressure vessel construction. All have complete stamped-in identification markings as shown. For protection, identification, and pleasing appearance, they are painted orange with non-toxic corrosion resistant paint.

American-made Welding Fittings

Designed and manufactured for utmost economy. No frills; just the solid value you would expect from **TAYLOR FORGE.**

T F

TRADITIONALLY DEPENDABLE

Compact line fittings are carried in stock by Taylor Forge Distributors in all important trading centers. The whole story is in Bulletin 619. *Ask for it.*

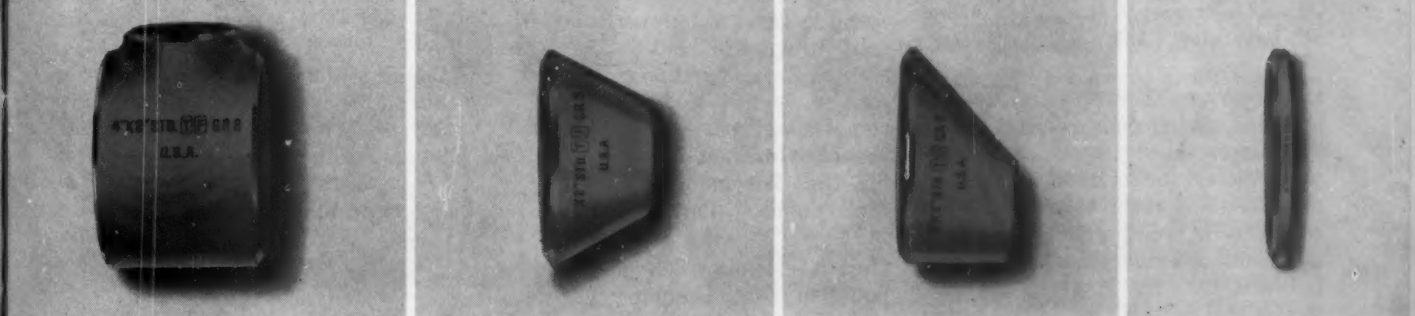
Taylor Forge & Pipe Works

General Offices and Works: P. O. Box 485, Chicago 90, Illinois

Plants: Carnegie, Pa.; Somerville, N.J.; Gary, Ind.; Houston, Texas; Fontana, Calif.; Hamilton, Ont., Canada

District Sales Offices: New York, Philadelphia, Pittsburgh, Atlanta, Chicago, Houston, Tulsa, Los Angeles, San Francisco, Seattle, Hamilton, Calgary, Montreal

REDUCING TEES	CONCENTRIC REDUCERS	ECCENTRIC REDUCERS	WELDING CAPS



COMPACT LINE *of Welding Fittings*

motive industries. Quick connect/disconnect is provided.

Ampeez units, utilized in major appliance equipment, are available in a choice of 7, 14 and 20-circuit configurations. Low insertion-extraction force, coupled with an amperage capacity to 25 amp, makes possible extensive flexibility of application.

Amp-Lok connectors, with a quick connect/disconnect feature, have now a broadened field of application.
AMP Inc., Eisenhower Blvd., Harrisburg, Pa.

Valves, Blenders, Control Units. Information is provided in Catalog 1161 on use of these air valves, blenders and high capacity acoustic terminal control units in air distribution systems. Air valves can have either manual or pneumatic controls and are used for velocity and pressure reduction, balancing in single-duct systems and control of static pressure unbalance in dual-duct systems.

Blenders are for control of velocity, pressure, temperature and mixing in dual-duct systems. Control of these factors and of volume and noise in dual-duct systems is provided by the ATC units.

Included in the catalog are rating charts, suggested applications and an attenuation analysis. Further information is provided by suggested specifications.
Carnes Corporation, Verona, Wisc.

Airborne Particle Monitoring. For continuous, unattended size, concentration analysis of airborne particulates, digital Model PC 200-A, a programmable instrument, is described in a flyer. The unit is available both with and without self-contained dilution facilities, essential for the monitoring of extremely polluted environments. Specifications detail 15 particle size ranges, starting with 0.32 micron; automatic size-range scanning arrangements; visual and recorded readouts and such details as physical and electrical characteristics.
Royco Instruments, Inc., 440 Olive St., Palo Alto, Calif.

Die-Cut Parts. Materials and services provided in custom die-cutting of gaskets, insulating parts and structural pieces for original equipment manufacturers and others in the air conditioning, refrigeration and heating fields are discussed in this new bulletin.
Beaverite Products, Inc., Die-Cutting Div, Beaver Falls, N. Y.

Positive Flow Boiler. Incorporated in the Powermaster positive flow, packaged, automatic boiler are such features as: side located furnace, a newly designed burner system, three-pass design, five sq ft of heating surface per boiler hp and standard dry-back or optional wet-back. Ratings, dimensions and approximate fuel consumption for models in sizes from 50 to 100 hp are included in Bulletin 1275.
Orr & Sembower, Inc., Reading, Pa.

Contactors and Starters. Descriptive of an extensive line of contactors and starters for air conditioning, refrigeration and resistance heating applications, 12-page Bulletin GEA-7316A contains information on new 20, 25 and 75-amp forms which were added to the line recently. Design features, including small size, front accessibility and straight-through wiring, are detailed. Engineering data, outline drawing and dimensions, selection guide and overload relay heater ratings are listed for all forms.
General Electric Company, Schenectady 5, N. Y.

High Temperature Wiring. Within the 29 sections of this 58-page catalog are synopses of government wire specifications, new product highlights and easy-reference tabulations on applicable military specifications. Ratings, physical dimensions, dielectric strength and temperature ratings are provided for the company's magnet wire, Teflon wrapped and extruded hook-up wire, miniature Teflon lead wire, Teflon tubing, cables, marking fluids, special jackets and other products.
Simplex Wire & Cable Company, Hitemp Wires Company Div, 1200 Shames Drive, Westbury, N. Y.

Piping Design. Use of a Univac Computing Center for piping design and analysis is described in a four-page bulletin. The computor program for flexibility analysis yields direct printout of forces, moments, stresses and deflections at each segment and terminal in the system under study. Based on tensor formulas and other relationships that hold true regardless of piping configuration, the program can be applied to both simple and complex systems.
Franklin Institute Laboratories, 20th St. and Benjamin Franklin Pkwy., Philadelphia 3, Pa.

Dust Collectors. Self-contained air pollution control equipment, developed for control and collection of industrial dusts, is outlined in four-page Bulletin 230. Illustrated and described are functions and applications of self-contained cyclone separators, fabric filters and combination fabric-cyclone separators in low volume systems.
John Wood Company, Air Pollution Control Div, Florham Park, N. J.

Infrared Heater Applications. Covering infrared heating in detail, 16-page Bulletin PE-2-61 describes such aspects as body heat loss and methods of controlling this loss. Two general methods for calculating requirements are explained. Seven tables are provided, including one that shows average outdoor winter temperatures for various parts of the country. Eleven graphs and engineering drawings are given.
Bettcher Manufacturing Corporation, Panelbloc Div, 3106 W. 61st St., Cleveland, Ohio.

Bearings. Covered in 12-page Bulletin 7104 are all sizes and styles of Life-Lube, permanently lubricated, ball-bearing pillow blocks, flange units and take-up units, as well as porous-bronze and babbitted pillow blocks. New Items listed are two-hole flange units, offered in shaft sizes of from ½ to 1-7/16 in. Included is a full page of babitted bearing information on load ratings and maximum shaft speeds.
T. B. Wood's Sons Company, Chambersburg, Pa.

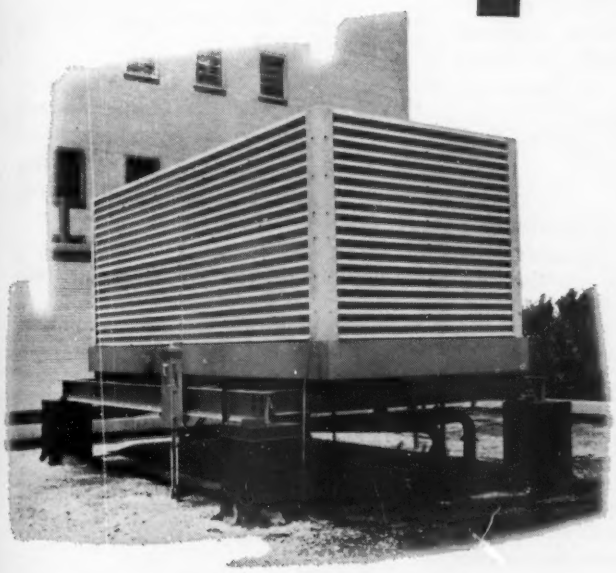
Cascade Control. Elements of cascade control of chemical processes in various fields are outlined in four-page Application Bulletin 91-53P-04. Several advantages are pointed out and various systems that achieve this type of control are presented schematically. Jacketed kettle temperature control and level control in a filtrate tank are cited as typical of applications in which cascade control can be beneficial.
Fischer & Porter Company, 26 Jacksonville Rd., Warminster, Pa.

Evaporative Coolers. Detailed information is provided in four-page Catalog 1961-4P for evaporative coolers for cooling jacket water, lube oil, compressed air and natural gas or for condensing steam and hydrocarbons. Various sections of the catalog cover design features, piping diagrams of sample units and dimensional drawings and tables.
Young Radiator Company, Racine, Wisconsin.

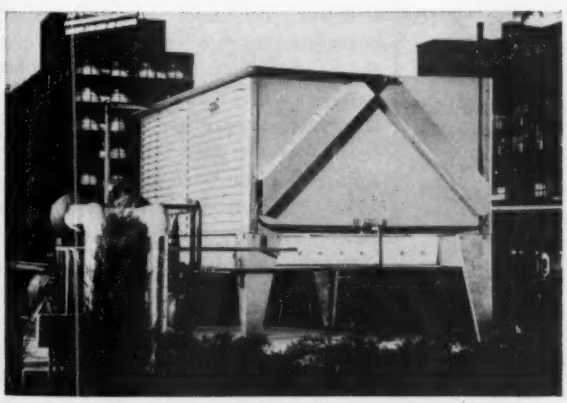
Gas-Fired Heaters. Descriptive of Series NB centrifugal blower-type, gas-fired unit heaters is four-page Bulletin A-917. Operating and construction features are detailed and an ex-

MARLEY DOUBLE-FLOW AQUATOWER at Sibley Memorial Hospital, Washington, D. C., provides cold water for large complete-building comfort cooling system at a low cost per gpm cooled.

In hospitals and wherever performance *is* paramount



MARLEY UNDERFLOW at Euclid-Glenville Hospital, Cleveland, combines minimum plan area, low height, high performance, compatibility and lowest operating and maintenance costs.



MARLEY DRICOOLER with winter control Damper automatically maintains uniform condensing temperature in year-around service at the research laboratory of a major agricultural products processor.

Pick any city (or cities), check the cooling systems of hospitals, laboratories, research facilities—other institutions where temperature control is critical—and you find that a broad majority have this in common:

MARLEY COOLING TOWERS

AND

AIR COOLED CONDENSERS

Why this widespread acceptance and approval of Marley products for the "chips down" jobs? Because they have in common three basic demands and Marley engineering fulfills them all, not just adequately . . . *impressively.*

Performance Is Paramount—Full performance from start-up throughout a long service life is fundamental in Marley design. Every component is individually researched and rated; the integration of known-capacity elements into each Marley product invariably produces the yardstick by which similar products are measured.

Durability Demonstrated—In thousands of installations Marley products daily (24 hours a day) prove the advantages of Marley structural and mechanical engineering that means rugged, rigid structures engineered to ignore corrosion, deterioration and vibration; that means fans and power transmission units designed and test-demonstrated specifically for cooling tower duty.

Maintenance Minimized—Inherent stamina of structure and mechanical equipment precludes major service demands; routine maintenance is simplified by complete accessibility to all product parts.

THE MARLEY COMPANY • KANSAS CITY 14, MISSOURI

ploded view photograph is utilized to illustrate the unit components. Performance data are tabulated and show heat input and output, air delivery data, blower pulley data and motor data for each of the eight sizes in the line. An outline type illustration provides all basic dimensional data needed for installation layout preparation.
American-Standard, Industrial Div, Detroit 32, Mich.

Pneumatic Receiver. Descriptive of the Series 670 Metagraphic pneumatic receiver, eight-page Bulletin A113.3.-3-2 supplements text with pictures and engineering drawings. Detailed are construction, design, size and capabilities of the instrument.
Bristol Company, Waterbury 20, Conn.

Refrigeration and Air Conditioning Products. Among items covered in 20-page Catalog 22 are accumulators, capacitor tester, door gasket notcher, fin-comb, gaskets, liquid leak detector, strainers, terminals and the following valves: access saddle, access T, adjustable line tap, can tap, capillary control, control for tapping valves, flappers, line tap, line port, Magni-Chek and spun copper Magni-Chek. Items are illustrated and dimensions are provided.
Watsco, Inc., 1020 E. 15th St., Hialeah, Fla.

Furnace Explosions. Analyzing the causes of 156 recent furnace explosions, 16-page Bulletin E-101 discusses the theory of explosions during flame-outs and presents practical recommendations for safe operating procedures.
Babcock & Wilcox Company, 161 E. 42nd St., New York 17, N. Y.

Regulators. Balance, pilot-operated and direct-acting valves—all pressure reducing regulators designed for reduction, regulation and maintenance of steam, water, gas or oil pressure—are discussed in eight-page Bulletin SM 71. Service data, connecting pipe sizes, construction specifications, applications and operation details are presented for each valve type. Charts list overall dimensions and weights of each valve in the line. Short descriptions also are given for reducing, regulating and block steel valves.
Strong Steam Specialties, 508 Sandusky St., Conneaut, Ohio.

Steam Heating Specialties. Included in eight-page Catalog 1501D are 56 illustrations and product drawings for this extensive line. Information is provided for capacities, dimensions and weights of radiator valves, inverted bucket traps, radiator traps, strainers, high pressure thermostatic traps, float and thermostatic traps and gauges.
Dunham-Bush, Inc., 179 South St., West Hartford 10, Conn.

Valve Sizing. "A practical Look at Valve Sizing" is a four-page discussion of what happens in undersized, oversized and correctly sized pressure reducing applications. A typical example is given, dangers of mis-sizing are indicated and illustrations are provided.
Richards Industries, Inc., Jordan Valve Div, 6013 Wiehe Rd., Cincinnati 37, Ohio.

Laboratory Controller. Basically a portable instrument for precise, automatic regulation of temperature, this unit is applied readily to measurement and control of pressure, vacuum and liquid level. Provided are proportional control (modulated duration pulse), on-off control, high limit alarm and an output signal for recorders. Bulletin 850, four pages.
Brooks Instrument Company, Inc., Hatfield, Pa.

Mounting Panels. For direct and belt-driven, propeller type exhaust fans, these mounting panels are the subject of Flyer 32. Dimensional information, diagrams and product illustrations are provided.
Air Conditioning Engineering Company (Canada) Ltd., 636 St. Paul St. W., Montreal 3, Que., Canada.

Electric Fan-Coil Controls. Features, electrical and installation specifications and applications for both thermostats and valves are covered in six-page Bulletin 54-3012. Also detailed are advantages of these controls when combined in a system.
Minneapolis - Honeywell Regulator Company, Commercial Div, 2753 Fourth Ave. S., Minneapolis 8, Minn.

Industrial Gas Data. Pocket-sized, 48-page Bulletin ADE-890 lists physical and chemical properties of oxygen, nitrogen, argon, helium, hydrogen, carbon dioxide, acetylene, neon, krypton and xenon. Similar information is provided on the medical gases ethylene, nitrous oxide and cyclopropane. Conversion tables provide a simple method for changing volumetric information from one equivalent unit of measure to another. Capacity data are given for industrial gas delivery and storage systems.
Air Reduction Company, Inc., Air Reduction Sales Company Div, 150 E. 42nd St., New York 17, N. Y.

Electrodes. Oil and gas burner igniter electrodes, glow plugs, flame rods, electric feed-throughs and liquid level control electrodes with Teflon or alumina insulation are described in four-page Bulletin OB6109. All are shown in dimensioned, cross-sectional drawings, and suggested applications, electrical and hydraulic characteristics are provided.
Auburn Spark Plug Company, Inc., 1099 York St., Auburn, N. Y.

Small Unit Heaters. Specification data on three types of small capacity steam and hot water unit heaters are contained in 20-page Catalog 46-106. Capacities of the heaters range from 13,400 to 1,062,000 Btu/hr and horizontal, vertical and four-way air discharge patterns are featured. Data provided include steam and hot water ratings, physical dimensions and electrical and piping diagrams.
Carrier Air Conditioning Company, Syracuse 1, N. Y.

Dehumidifier. Manufactured in several sizes, the HoneyCombe dehumidifier utilizes a new permanent desiccant structure pierced with corrugated flutes in the form of a slowly rotating disc which passes progressively through an air stream which it dries, and a heated air stream to which it gives up moisture. Longitudinal flute design is cited as preventing internal bypassing within the wheel between the drying and regeneration air circuits. Descriptive of the unit is four-page Bulletin 3003.
Cargocaire Engineering Corporation, 15 Park Row, New York 38, N. Y.

Heating Products. Three main sections divide 20-page Catalog H-61: oil burner nozzles, oil burner accessories and draft controls. Product illustrations, construction and design features, technical information and specifications are provided. Given in the oil burner nozzle section are gph capacity and spray angle charts, illustrated spray patterns and engineering recommendations for proper application of each type.
Wm. Steinen Manufacturing Company, Heating Div, 47 Bruen St., Newark 5, N. J.

Unit Heaters. General design and engineering specifications, steam and hot water rating tables and air distribution information for downflow unit heaters are included in eight-page Catalog 765-2. Units are illustrated.
McQuay, Inc., 1600 Broadway N.E., Minneapolis 13, Minn.

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Comment

ASHRAE JOURNAL

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AS TO MATTERS OF MARK-UP

If a manufactured product can not be sold, it might better never have been designed and fabricated.

No engineer can separate his work or his rewards from the final assault upon the consumer's pocketbook.

Let us assume an impossibly ideal product deserving of equally unlikely wholehearted acceptance to the buyer. There still remains the gap that only selling and promotion can bridge.

The trouble is that instead of being a single step it may become a whole flight of stairs where each and every level of distribution expects to find its own justification for existence and its own profit. In doing so, each level cites its supposedly indispensable contributions to the overall economy and adds inevitably to the cost to the consumers.

We think that the only truly acceptable system of distribution is that which results in the least cost to the purchaser — measured, of course, in terms of all properly recognizable cost values and not alone in the mere dollar spread.

Far from being an academic matter, the amazingly accelerated growth of discount houses but points the matter up. On the Pacific Coast, one of the largest of automobile makers is under charges for its alleged refusal or reluctance to have the distributor-dealer set-up bypassed. So-called Fair Trading has been in defensive retreat for years on many fronts.

So what is to be done about it?

What should be done about it?

Our answer to these questions is "Nothing".

NOTHING SACRED?

What has so often been spoken of, in another connotation, as the bloodless verdict of the market-place might well be permitted to prevail. If one level in the distribution stairway is unjustified it might well be eliminated. For, uncomfortable or no for the parties involved, various uneconomic things have a way of disappearing.

What has happened to the producers of prairie wagons, to the whip makers, to the manufacturers of oil lanterns, of music boxes, of magic lanterns and of spinning wheels? Are distribution methods any less likely to be proven outmoded and supplanted by better ways?

Edward R Searles
Editor

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Successive developments that led to the

Vapor Cycle System

for the Boeing 707

Air conditioning of Series 707 airplanes is with either an air cycle or vapor cycle cooling system. The former was chosen as the basic one because of its light weight and high state of previous development in the KC-135 airplanes. The vapor cycle system was offered to those customers who desired ground cooling and heating with external electrical power only, thus eliminating the necessity of using a ground air-conditioning truck. This approach had two main consequences. First, the vapor cycle system was compromised by early design decisions which accommodated the basic system, and secondly, this all-new system required a concentrated engineering effort in order to develop it in the available time.

The vapor cycle system, to date, has been chosen by four different airlines. In all, more than 100 of the 707 and 720 airplanes using it have been delivered or are on order. The installation of the vapor cycle equipment, including wiring, increased alternator capacity, etc., has resulted in a weight penalty over the air cycle system. The primary reason for the selection of an electrically powered refrigeration system was to provide ground cooling with a simple electrical plug-in from the ramp or from a ground electrical cart. The increased operational efficiency of the vapor cycle system redeems the effects of the extra weight with respect to airplane fuel limitation. The main drawback of this system in its present state is its control complexity fostering a reliability which is somewhat lower than the relatively simpler air cycle system, even though each of the individual components is reliable.

E. J. OVERMYER
Member ASHRAE

The system as originally delivered is shown schematically in Fig. 1. The design was for a capacity of 10 ton at an ambient temperature of 100 F with recirculated cabin air at 125 lb/min across the evaporator. Certain other in-flight cooling requirements were imposed but this appeared to be the one which would determine the size of the components.

The space allotted to the cooling equipment was of such a shape and size that the installation problems became quite difficult. An objective which complicated the installation was that the refrigeration circuit be sealed, and installed and removed as a unit. This objective was realized and has subsequently proven to be well worthwhile.

There were also performance compromises imposed by the space limitations. Air side pressure loss, which will be discussed in more detail later, became high. Maximum efficiency heat exchangers were required. The manufacturer therefore decided to use a semi-flooded type evaporator, since the flooded surface could be more compact. This in turn led to more operational problems, since a flooded system is more prone to slugging difficulties than is a dry expansion system.

CONTROLS WERE ONE PROBLEM

The refrigerant side controls utilized in the first production system were the source of much development activity. Referring to Fig. 1, the evaporator pressure regulator was used to modulate the cooling capacity. Raising or lowering the pressure at which the refrigerant

Vapor cycle system eliminated the ground air-conditioning truck

Refrigerant side controls led to much developmental activity

Performance compromises were imposed by space limitations

Ram air pressure fluctuations were a problem, too

Air-to-air precooler for flexibility, increased reliability

E. J. Overmyer is a Research Engineer with the Boeing Company. His previous, related paper was "How face air velocity affects airplane comfort" as published in the August 1961 JOURNAL.

was evaporated correspondingly raised or lowered the evaporating temperature and hence varied the amount of heat removed from the air. Quite briefly, this valve was positioned through a balance of refrigerant pressures and spring and bellows forces. This balance, and hence valve position, was effected by movement of an electrical actuator which received signals from the temperature control system. The response of this valve was sluggish and non-linear. It went through a number of minor changes, most of which contributed only minor improvements to its operations.

The expansion valve used was of the conventional thermostatic type. Theoretically, a constant superheat should have been held at the evaporator discharge, assuring that the refrigerant was completely evaporated and that no liquid slugs would be carried into the compressor inlet. The primary deleterious effect of liquid slugging in a centrifugal compressor is evinced by surges of compressor motor current. The difficulty was two-fold: one, locating the power bulb such that a representative refrigerant condition would be sensed at all times, and two, obtaining a response rapid enough to prevent slugging under transient load conditions. The problem was complicated by the necessity of keeping the amount of superheat control fairly low (10 F was the objective) in order to obtain the desired capacity at design full load conditions.

The third control in the loop was the compressor surge bypass valve, its function, of course, being to keep the compressor from surging when the refrigerant flow was throttled by either or both of the other valves. This was done by sensing total and static pressures at the compressor discharge and using these signals to position a valve maintaining the refrigerant velocity at a minimum value. Transient response of this valve was slow and the adjustment required to keep the valve closed at design operating conditions, yet allow it to open when necessary to keep the compressor out of surge, was quite tricky.

The stability of these three controls when combined in the system was extremely poor. Each time one of them moved the other two had to adjust, which in turn caused the first one to move again, and so on.

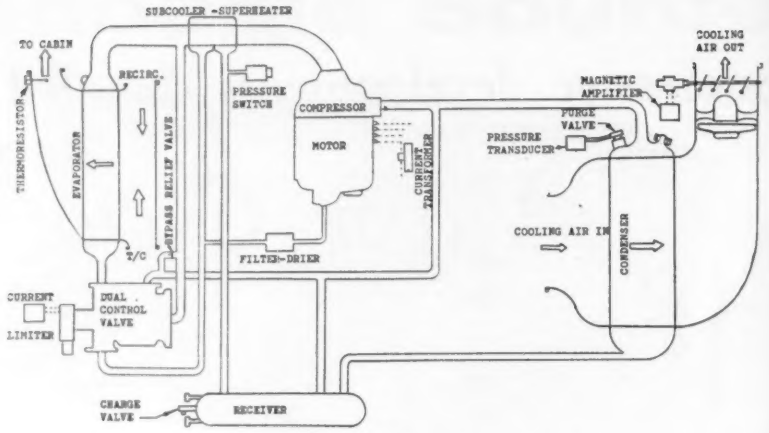
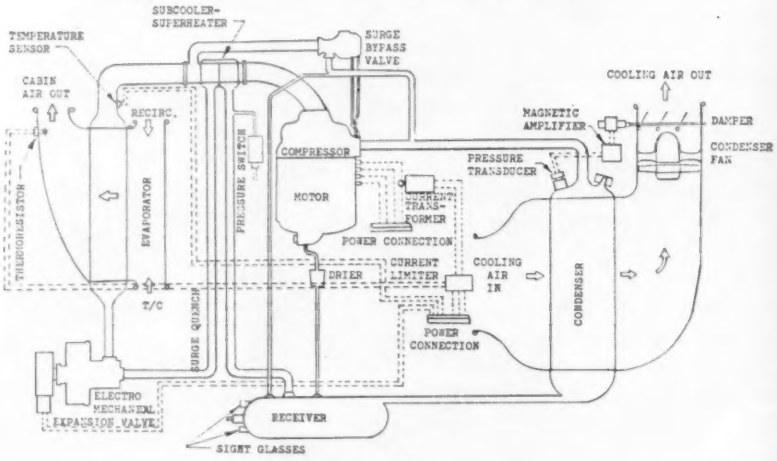


Fig. 1 Original refrigeration circuit

Fig. 2 Amended circuit as installed in first plane



The three basic components in the refrigerant loop, i.e. compressor, condenser and evaporator, have been comparatively trouble free, and have gone virtually unchanged since the first production system. The early difficulties encountered with the evaporator and condenser were the common ones of proper refrigerant and air distribution, and in passing it should be mentioned that refrigerant distribution in the evaporator is accomplished by means of a high pressure drop spray tube.

The first major change in the system is shown schematically in Fig. 2 and consisted primarily of a change in the control concept and the addition of a subcooler-superheater between the evaporator discharge and compressor inlet. The latter item improves the cycle efficiency slightly and helps to assure the complete evaporation of refrigerant. The evaporator pressure regulator was deleted and the thermostatic expansion valve was replaced with an electrically actuated valve, positioned by signals from the temperature control system. The capacity then was controlled by throttling the flow of refrigerant at the inlet to the evaporator.

It was found desirable at this point to add also a current limiter. This consisted of a current transformer on one of the compressor motor leads, which fed a signal into a magnetic amplifier. This then affected the signals from the temperature controller, either letting cool signals through, blocking them off, or changing the cooling signal to a heating signal, depending upon the magnitude of the compressor motor current. Later on,

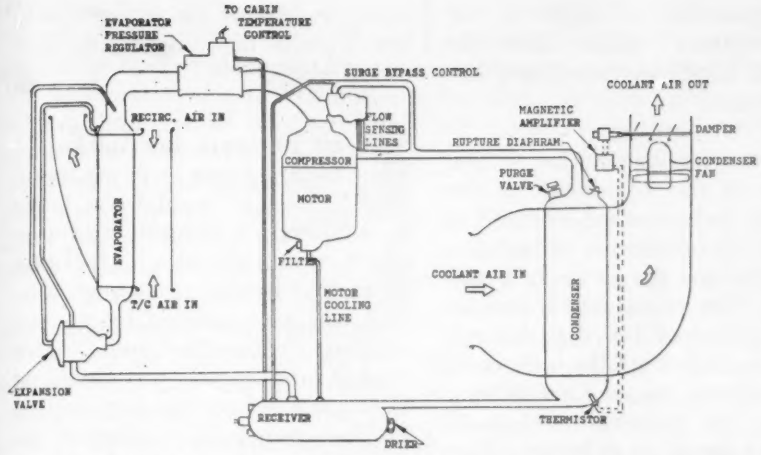


Fig. 3 Presently specified refrigeration circuit for vapor cycle system

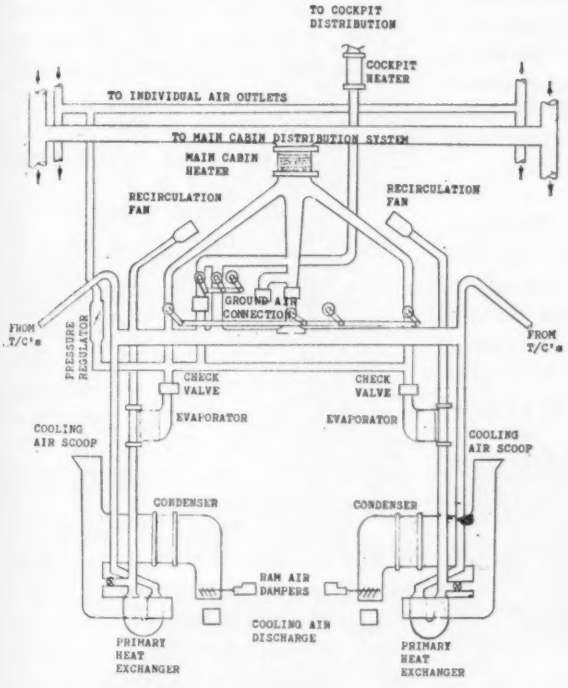


Fig. 4 Complete schematic for vapor cycle system

evaporator discharge air temperature bias was added to prevent liquid carry-over to the compressor when the evaporator was loaded lightly.

The response and stability of the system were now both much improved. However, the compressor surge control continued to be a problem.

IN THE FINAL DESIGN

As presently used, the system is shown schematically in Fig. 3. The major change here was the combination of the functions of surge bypass valve and expansion valve into a single article, which now is referred to as a dual control valve. This was deemed advisable after the units had been in service for about a year and continued to be plagued by surge control problems. The new control has the two valves on a single shaft so that when the expansion valve is closed, the surge bypass valve is opened by the same actuator, and vice versa.

Condensing pressure control and condenser fan damper – This control is discussed separately because its function is not associated closely with that of the other refrigerant circuit controls, and because the basic problem with it arose some six months after the decision to change to the system shown in Fig. 2 had been made. The original reason behind incorporation of this control was to provide a means of reducing ram air drag when the full potential of the ram air heat sink was not required. There was another equally important reason for modulating cooling airflow, although the people involved in this program did not become fully aware of all the implications until after flight testing of the first systems had commenced. At that time it became apparent that excess cooling airflow (at the extremely low ambient temperature encountered at high altitude) was causing a condensing pressure so low that the refrigerant would not flow across the expansion valve-distribution tube restriction. This resulted in zero cooling which in turn prevented the condensing pressure from building up. The compressor motor refrigerant flow also dropped off, and although the motor load was quite low, the compressor would eventually be shut off due to overheat.

The excess cooling airflow resulted from fixed leakage that had been designed into the damper in order to provide condenser fan cooling in cases where the fan would be operating with the compressor off. A re-design of the dampers to make them tight-sealing fixed the problem of low condensing pressure.

The sensing device was changed from a condenser discharge air temperature probe to a condensing pressure transducer in order to provide faster response. Signals from this transducer go to a magnetic amplifier which controls an actuator, positioning the damper to maintain a condensing pressure of approximately 60 psig. The control circuitry was rigged to cause the dampers to go wide open if the airplane is on the ground and if the compressor is not operating. This assures that the fan will have cooling air if the compressor trips off on overload, for example.

Compressor and motor—The lubrication of the compressor and motor ball bearings is revolutionary in that there is no oil sump. Approximately 5% of oil by weight is added to the refrigerant. Since oil and Refrigerant 114 are miscible, part of the oil will exist as a mist whenever the compressor is operating, and will be circulated through the system. It is estimated that about 1% by weight of oil is present in the motor cooling refrigerant which

is circulated through the bearings. The motor cooling flow is about 5 lb/min of Refrigerant 114, of which only part goes through the bearings, thus a conspicuously small rate of oil flow is required for satisfactory lubrication.

The motor cooling refrigerant flow on the early systems was somewhat marginal under overload. It could not be increased to any great extent without compromising the capacity or adding complication to the system. The cooling flow is metered by a fixed restriction. This could have been changed to a variable restriction, thermostatic device, to handle the extra motor heat at overload, or the restriction to flow could have been reduced, which in the present circuitry would have taken extra refrigerant from the evaporator at all times. In the system shown in Fig. 3 the take-off point for the motor cooling refrigerant was changed from the receiver to a point downstream of the subcooler. This increased the effective cooling of each pound of refrigerant and appears to have been an appreciable improvement.

Safety devices – As originally conceived, the only protection devices in the system were a compressor motor thermostat and a high pressure rupture diaphragm in the condenser. After some early laboratory failures of the motor, in which the case was burned through by electrical arcs, it became apparent that this protection was not sufficient.

The thermostat was changed to a current-biased motor overload protector for faster, more positive response, and a motor fault detector was added. This latter device consists of running the four wires to the compressor motor through the eye of a current transformer and sensing the secondary output. As long as the motor is operating normally the secondary current is essentially zero. If a fault occurs, a current is generated in the transformer which actuates a lockout relay. The re-set switch for the lockout relay is located in the equipment bay and thus can be operated on the ground only.

A high condensing pressure switch also is incorporated. This and the overload protector can be re-set manually in flight at the flight engineer's panel after the overload producing condition has been cleared.

Evaporator air distribution – The location of the equipment in the wing to body fairing dictated a fore and aft orientation of both the evaporator and the air ducts to and from it. On recirculation the air makes a turn of 180 deg through the evaporator. With the turbocompressor source the air turns 90 deg entering the evaporator, then 90 deg back again as it leaves. This presented a difficult air distribution problem in order to obtain maximum heat exchanger efficiency, and it added pressure drop, which is wasteful of refrigeration capacity. The problem was alleviated partially by the installation of turning vanes in the inlet header.

Electrical control system—The electrical controls which will be discussed here are those external to the refrigeration pack and not inclusive of the temperature control system components. In the beginning the general objective was to provide controls which would demand a minimum of attention from the crew. In this respect, simple on-off switches were provided, one for each pack, and monitoring of the heavy starting loads of the various components was accomplished by means of timers and relays.

Operational difficulties which have since arisen have resulted in further complication. The electrical loads became excessive, so that an interlock between the refrigerant compressors and other airplane equipment was necessary. A stray circuit from a radio noise filter on the temperature regulator was discovered, which at times prevented the expansion valves from opening. This was solved by the addition of anti-feedback relays. It was found that excessively long compressor starting times were occurring due to liquid slugging, so means were provided to close the expansion valve and pump the evaporator dry prior to shutdown. Difficulties with the temperature control system led to an over-ride switch and to a temperature control (air mix) valve switch. A condenser fan switch to permit an increase in cooling air at low speed, low altitude flight conditions was added also.

Ram air pressure fluctuations – When the first vapor cycle equipped 707-320 Intercontinental was flown it exhibited a definite vibration which caused the crew to feel that the cockpit was bouncing vertically. This was accompanied by a characteristic "choo-choo" noise. This trouble subsequently was identified as emanating from the ram cooling air ducts, when the condenser fan damper was closed or nearly closed. At this time the scoop opening was too big for the increased downstream restriction and alternate stalling and recovery of the scoop occurred. This resulted in a pressure fluctuation which was felt and heard in the cabin, and which damaged the cooling air ducts.

After an extensive laboratory and flight test program, a fix was devised which consisted of a 20 sq in. hole in the aft end of the ram air duct. This "dump" hole was ducted overboard. Tests showed that the condenser and primary cooling airflow was not adversely affected, apparently because the extra flow increased the scoop recovery by a compensating amount.

When the first 720 was flown it was discovered that it, too, was subject to the phenomena of choo-choo and vertical bounce under the same conditions as the Intercontinental although not as pronounced, whereas this had not been noticed on the 707-120. The only difference between the 707-120 and the 720 was the addition of a wing leading edge "glove" ahead of the scoop, which would increase the boundary layer effects slightly. On this airplane a combination of a slatted lower inlet lip on the scoop and a 20 sq in. dump hole gave acceptable damping, and the air conditioning system performance did not deteriorate with condenser fans operating at low altitude. However, some of our future airplanes will be equipped with variable area inlets to reduce the drag as well as preventing the ram air pressure fluctuation.

Recirculation Fans – The performance of the original recirculation fans, which provide air to the

evaporator on the ground, met our specification requirements. However, the overall pressure drop of the evaporator, ducts, distribution system and return air path exceeded estimates. When we started producing airplanes we discovered that most fans were producing less than design weight flow, and in some cases were operating in the unstable flow region. This led to the rather expensive procurement of fans of an almost entirely new design. It also led to wasting refrigeration to remove the fan energy added to the air. This points up the need for generous-sized ducting and advantageous location of the refrigeration equipment, but in any airplane system it is natural that there be pressure towards light weight and small size.

Vapor cycle system programmers— In the air cycle system all temperature control is accomplished by positioning a multi-unit gang valve. Signals from the temperature regulator go directly to a single actuator to cause the valves to move either in the direction of increased cooling or increased heating. In the vapor cycle system there are four units which affect cabin temperature, namely the left and right hand refrigeration packs, a three-way air mixing valve, and an electric heater.

Because of this, and the fact that the control system had evolved partially prior to the time that attention was given to vapor cycle control, a programmer was added which is motor-driven by signals from the regulator. The programmer has a series of cam switches on a shaft which open or close the proper circuits to direct cooling or heating signals to the four aforementioned units.

In the airplanes equipped with vapor cycle units, the programmers were at first the source of many of the air conditioning malfunctions. These were due to failures of relays within the programmers which caused them to drive to either full cooling or full heating. The programmer had been designed so that both a "manual" and automatic motor could drive the shaft through a differential gear. In the case of the relay failures this did no good since both automatic and manual control were lost. The programmer was redesigned to separate the manual and automatic circuits, and higher amperage rating relays were installed. Presently, the programmers seem to be performing adequately.

Conditioned air mixing valves, main cabin — The assembly consists of three valves, a master (hot) valve and two slave (cold) valves. The hot valve is completely closed when the cold valves are wide open and vice versa, with a linear relationship between the two extremes. The control system is such that the valves are driven to the full cold position on the ground so that the minimum restriction is imposed on the recirculation fans. On the ground, supplemental heat is supplied by the heaters only. In flight, the valves are full cold as long as the main cabin requires any vapor cycle refrigeration. As less cooling is required, the valves will modulate. After reaching the full hot position, an electric heater will operate. The system is shown schematically in Fig. 4.

Primary heat exchanger—There are two basic reasons for using an air-to-air precooler or primary heat exchanger in series with a vapor cycle evaporator. First, the low altitude flight cooling requirement may exceed the ground cooling requirement, and the PHX may properly be utilized to reduce installed equipment weight. Secondly, the air-to-air heat exchanger is extremely reliable. If there is a malfunction in the mechanical refrigeration system all cooling is not lost, and bearable cabin temperatures may still be obtained. Because of this second reason we are presently procuring larger primary heat exchangers for installation in later airplanes. This is intended to reduce the operating time for the vapor cycle equipment as well, by this means increasing its reliability.

There is, of course, a limit to the size of the primary heat exchanger, otherwise we would be carrying the heavy vapor cycle refrigeration equipment for ground and low altitude use only. Actually, the mechanical refrigeration equipment is more economical to operate because it rejects heat at a higher level, thereby using less ram cooling air.

Electric heaters — In line with the concept of an electric powered self-contained air conditioning system, electric heaters for the main and control cabin were installed. These have a total capacity of 30 kw, 3 kw for the control cabin heater and 27 kw for the main cabin heater. A temperature rise of 28 F is obtained which is coupled with the fan rise.

The heaters are an extended surface, plate-fin type with sprayed-on ceramic-coated heating elements. The air pressure drop through them is low when clean, however, the fins do collect dust and must therefore be cleaned periodically.

CONCLUSION

The 707 vapor cycle air conditioning system, like any all-new system, has had its share of development problems. Some of these problems were common to those encountered in industrial applications, but many of the problems were complicated by requirements peculiar to an airborne system. Many of the solutions which we might have used were not available to us because of adverse effects which they would have had on the rest of the airplane. We feel that we now have a good, reliable system, but we are working steadily to effect further improvements.

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Discussion of

Some Strength Characteristics of Ice

at the interface

J. K. STENE

W. E. FONTAINE
Member ASHRAE

Several research programs to determine the adhesion strength of ice formed on various surfaces have been reported upon previously. The strength has been calculated based on the average stress required to separate the ice and the surface in either tension or shear.

Sibbitt, Fontaine and Dotson[11] conducted a series of tests to determine relative tensile and shear stress when ice adhered to various metal surfaces. The tests were conducted in a large cold room which was maintained at a temperature of about 20 F. Distilled water was used to eliminate any variables present in tap water. Care was used to apply the load evenly as their preliminary tests showed that the stress was a function of the rate of loading.

In these investigations when ice was formed between two cold metal surfaces, the ice failed at a crystal interface. This was attributed to ice formation advancing on fronts from both surfaces. When one surface was left warm, this type failure was eliminated. However, the breaking plane in the tensile tests always occurred in the ice, but the breaks moved closer to the surface that was originally cold.

Test results in tension were lumped together for all clean metal surfaces. The metals used were brass, copper, aluminum, and steel. The average tensile strength was given as about 105 psi for ice greater than one-eighth in. thick. When tests were performed varying the thickness of the ice, it was found that the stress increased as the thickness decreased.

In another series of tests they treated some metal surfaces with silicone compounds. The average value of the tensile stress for all tests was cited as 8 psi. There was noted a tendency for increasing stress as the number of removals from one surface was increased. This was attributed to removal of the silicone surface with the ice. In this series complete removal of the ice was achieved. In the shear tests the shearing strength of adhesion was determined to be less than the tensile strength of adhesion, averaging about 65 psi, and all separations occurred at the interface and not in the ice.

Results of these tests are illustrated with bar graphs which show the range of stress values for the tension tests and for the shear tests. There is a bar for each class of surface. The frequency of any value is not illustrated, nor is the spread within a range shown.

Jellinek[6] conducted extensive tests on the variation of the tensile stress with variation in geometry. The ice specimens were cylindrical in geometry. The interface was made by freezing the ends of these ice cylinders to stainless steel disks. Water-ice was formed from water that had been prepared by passing through an exchange resin to free it of electrolytes and by boiling to remove air. Snow-ice was formed by sieving snow into water and then freezing the mixture. The metal surfaces used were kept in benzene and cleaned in benzene before the tests.

The tests conducted were at constant temperature and were to measure the tensile strength as a function of the rate of loading, the thickness and the cross-sectional area of the specimens. The results were formulated in terms of the cross-sectional area and the volume. At −4.5 C the following equation applied "over a thousand-fold range of volumes":[6]

$$S = (2.74\,AV^{-0.84} + 9.4)\ \text{kg/cm}^2$$

S = tensile strength
A = cross-sectional area in cm^2
V = volume in cm^3

In the above tests the breaks were partly adhesional (at the interface) and partly cohesional (in the ice). None of the tests reported was for

W. E. Fontaine is Professor of Mechanical Engineering and J. K. Stene is an Associate Professor of Mechanical Engineering at Purdue University. This paper was presented at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

clean separation between the ice and metal. These tests also showed a tendency reported by other investigators; that is, for increasing stress with decreasing thickness. The effect of the rate of loading was that an increase in rate was found to decrease the proportion of the break that was adhesional.

Raraty and Tabor[10] froze ice in a mold so that torque could be applied to a metal cylinder frozen along the axis of the ice cylinder. For the tests torque was applied to the metal cylinder. The mold held the ice stationary so that the failure would occur at the cylindrical interface between the ice and the metal cylinder. Sometimes the failure was in the ice itself.

In their experiments the stainless steel surfaces were prepared by polishing, degreasing, cleaning in analar acetone and air drying. At first they boiled the distilled water to liberate the absorbed air, but when it was noticed that air bubbles tended to be trapped away from the interface this was stopped.

As torque was applied in these torsional shear tests, the investigators found an initial yield was dependent on time. When this variable was eliminated by rapid load application, there were wider variations in the torque values at failure. If the torque was applied slowly and at a uniform rate, there was less scatter in the results. The shear stress at failure was found to be dependent on temperature and increasing almost linearly as the temperature decreased.

Two metals were used in these tests. The rotated cylinder was of stainless steel for the tests shown graphically by the authors, and aluminum was said to have been tested. Less stress was said to be required for failure when stainless steel was used than when aluminum was used. The failure was always in the ice when aluminum was used.

These investigators ran another series of tests in which nonmetallic substances were used for the rotated cylinder. For the materials used it was found that the stress values increased as the temperature decreased in the range from 0 C to approximately –10 C. In the range from –10 to –30 C the stress value remained essentially constant. A plot of shear stress versus temperature showed that each material had a curve different from each of the other materials. Some further tests using metals covered with thin plastics films gave results similar to the solid plastics and not like the solid metal.

Stein[12] carried out two series of tests. One was a set of tension tests; the other a set of shear tests. These tests were conducted in a cold room where the ice and the environment temperatures varied between 18-22 F. Careful control was kept over the water condition and temperature, and cleanliness was watched carefully in regard to the solid surfaces.

Results of the tension tests showed a large spread in the stress values at separation. This spread was especially large where the break was entirely within the ice. For adhesion to metals, some stress values were over 300 psi; whereas, for adhesion to non-metallic surfaces all values were under 150 psi.

The shear tests showed less deviation from an average value, but some slow sliding of the ice over the surface was observed. All stress values regardless of surface were below 150 psi. Several different metals and some non-metallic surfaces were used in the tests. The metals showed the greater strength of adhesion. Complete removal of the ice was achieved with the shear tests. Extensive tables and graphs are presented to show stress values for each surface and each test.

Several general statements have been cited regarding the difficulties involved in conducting adhesion tests. In one article[10] three basic difficulties were noted. The first concerned the necessity for reproducible cleanliness of the surface. The second concerned the difficulty of applying a uniform load at the interface when working with tension tests. The third difficulty concerned the pattern of stresses at the interface. "The greatest uncertainties arise from the pattern of stresses generated at the interface. In many cases these are very complex and discontinuities may produce large stress concentrations of unknown magnitude. In addition, the very act of forming the interface may leave residual stresses that will play a part in the observed adhesive strength. This occurs for example when clean metals are pressed together; there is evidence to show that strong junctions are formed at the interface although in general little normal adhesion is observed. This is because released elastic stresses pull the junctions apart one by one as the load is removed."[10] Other comments pertinent to conducting adhesion tests are summarized below.

Two generalizations are that the area of real contact is important in determining adhesion forces, and that there are no smooth surfaces since all carefully polished surfaces have peaks and depressions which are large compared with molecular dimensions.[4] A conclusion that can be drawn is that the roughness of a surface can

Table I Percent Deviation of Equation from Average Shear Stress Data

Surface Material	Temp R	Stress From Equation psi	Avg. Stress minus Calc. Stress psi	Deviation %
Anodized Aluminum	461	350.7	6.0	1.7
	462	333.8	—10.6	— 3.3
	463	317.8	3.6	1.1
	464	302.6	0.9	0.3
Stainless Steel	462	246.8	1.5	0.6
	463	242.7	— 1.4	— 0.5
	464	238.7	— 0.8	— 0.3
	465	234.8	1.0	0.4
	466	231.0	0.3	0.1
White Teflon	461	73.3	1.8	2.4
	462	70.6	— 2.4	— 3.5
	464	65.5	0.8	1.2
Black Teflon	461	74.5	17.9	24.0
	462	62.9	—17.9	—39.8
	463	53.3	1.7	3.1
	464	45.0	4.4	8.9

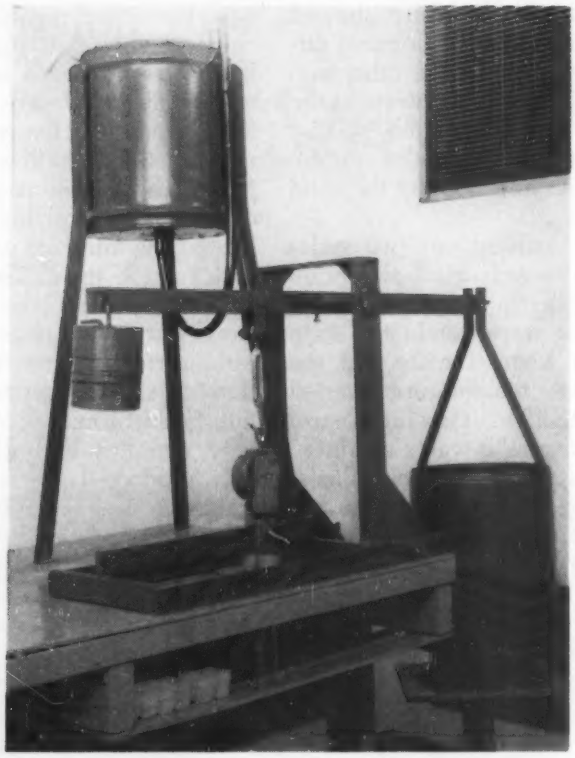

Fig. 1 Lever system at the beginning of the test. Ice is frozen to a vertically positioned surface

cause strong and weak points in the interface due to the adhesion forces of near points.

Contamination also is seen to be a real factor in any tests. This is true since contamination reduces the area of real contact. Rough surfaces can be contaminated even when handled carefully. Absorbed gas films, especially water vapor and carbon dioxide, prevent complete contact between the two solids and thus affect the adhesion strength of a junction between them.[4]

The physical position of the ice in an adhesion strength test places the ice in the position of an adhesive and the tested surface in the position of an adherent. Comments on this view of the problem are worth considering, as careful control measures for an experiment can be deduced from such discussion.

In a discussion by Bikerman[2] on adhesive joints, it is indicated that when an adhesive is applied as a liquid the rate of solidification determines the crystal size and the strength of the solid adhesive. At the same time, the thinner the adhesive the stronger the joint, since there is a greater probability of weak spots in larger volumes. These weak spots make stress distribution in larger volumes less favorable than in small volumes. The greater strength of smaller volumes has been well documented in the work of Jellinek[6] and was well known to other investigators.

Raraty and Tabor[10] write of the stress problem involved when ice freezes at 32 F and then is cooled further. There is a difference in the coefficients of contraction of ice and of the solids to which it is frozen. Metals have smaller coefficients than ice so that tensile stresses are set up in the ice, parallel to the interface. Likewise, compressive stresses are produced in the ice when ice is frozen to plastics which have greater coefficients of contraction than the ice.

Ingersoll, et al,[5] discuss the orientation of crystals when water freezes. The orientation being that the long axis is normal to the cold surface to which it is freezing. Truby[13] studied ice with an electron microscope to ascertain a relationship between the structure of ice and the rate of growth of the ice. All his samples indicated a micro structure of coordinated hexagonal prisms in bundles. The size of a bundle varied from one-half to 20 micron in width and from one to ten micron in length. The microscope pictures failed to show any relation between the size distribution and the rate of growth. Observation of the growing surface of the freezing ice showed pyramidal pits with hexagonal bases.

Bjerrum's[3] discussion of ice structure finds ice to be plastic in behavior because of easy slippage along hexagonal layers of the crystal. Nakaya,[8] in extensive tests on the formation of Tyndall flowers (voids in the ice due to internal melting caused by radiation), found that internal melting followed by refreezing left voids in the ice which were filled with water vapor. He determined that milky layers in ice were layers of these tiny cavities. Under these conditions there are indeterminate internal stresses.

All tests conducted in the strength of adhesion experiments can be summarized qualitatively by saying that some organic surfaces are effective in the release of ice, whereas most inorganic surfaces are ineffective.

In the report of Berghausen, et al,[1] it is stated that "It seems unlikely, ... , that any wholly inorganic coating or material of the type of most metal oxides or salts will ever be found such that the adhesive bond to ice is weaker than the cohesive strength of ice. Very probably the only types of coatings of surface treatments which will lead to ready release of ice from surfaces are those in which a surface of covalently-bound atoms is presented to the ice." This statement covers all results noted above, especially the results that accrued from tension tests.

Table II Constants for Each Surface in Equation

$$S = c(1 - dt + et^2 - ft^3)$$

Surface	c	d x 10^1	e x 10^3	f x 10^4
Anodized Aluminum	368.4	4.95	1.3	0.0
Stainless Steel	255.1	1.66	0.2	0.0
White Teflon	76.1	3.78	0.8	0.0
Black Teflon	88.1	16.86	14.4	8.0

EXPERIMENTAL DISCUSSION

The test apparatus and equipment used was that prepared by Stein[12] with some modifications. The shear puller shoulders were widened so that after separation, the corners of the puller were not dragged across the face of the tested surface. The mold shape was changed from square to round, and larger holes for air escape were made in the mold. The can containing shot had a conical bottom placed inside to improve the feed of the shot. Tare weights were made to eliminate one of the weighing steps.

Fig. 1 shows the ice frozen to a vertically positioned surface. The pull was applied in a vertical direction by a simple lever system. With the tare in the weighing bucket and the apparatus in place, the lever arm was balanced. The force necessary for separation was measured as the amount of shot added to the bucket to cause separation. This value was adjusted for the lever arm ratios. The rate of flow of the shot into the bucket was checked to be sure that uniform rates of force application applied to all tests.

Preliminary tests were conducted to establish a uniform method of preparing and handling the water and for cleaning and preparing the equipment. It was felt that this procedure gave as uniform results as possible.

The most uniform ice formation apparently resulted when the holder and the surface were precooled to the test temperature, the puller and the mold inserted at room temperature and the water poured into the molds at some chilled temperature. Throughout this experiment the water was kept in a refrigerator at 40 F up to the time of pouring. This procedure allowed the ice to freeze with crystals normal to the surface and to reach the coldest temperature in the same time for each test. No imperfections were noticed at the interface when this procedure was followed.

No large change in stress was expected over the range of temperatures reported here. The stress values at higher temperatures were considerably lower, but failure of the interface exhibited definite sliding of the ice over the surface before the sudden release which was looked for as the end of the test. This sliding was visible on the apparatus during the test and also on the test surface after separation.

Failure of the joint with sliding did not agree with the definition of failure followed in this discussion and therefore was eliminated. Failure following sliding indicates that failure is a function of the loading rate. Several checks were made on this point by stopping the loading as soon as sliding was observed. Sliding continued until failure occurred. The total load was less than would have been applied had the load been added continuously during the period while sliding occurred and before the ice left the surface. The sliding did not stop when the loading stopped.

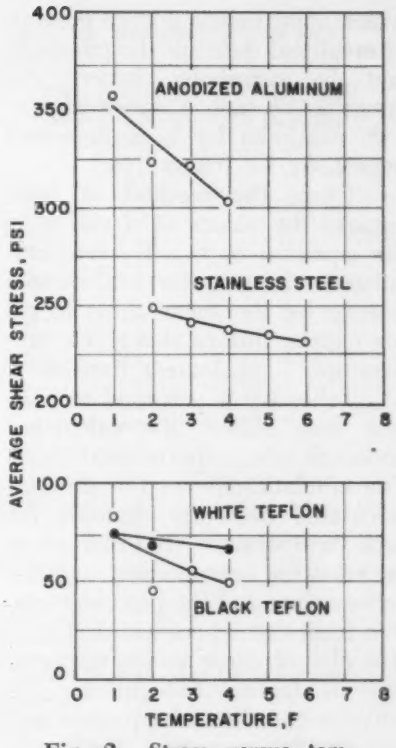

Fig. 2 Stress versus temperature

THEORETICAL DEVELOPMENT

When the data from the experiments were plotted on coordinates of average shear stress against temperature for each surface, it was observed that there was a definite tendency for the average shear stress values to decrease as the temperature increased. This suggested the use of a hyperbolic curve to represent the data since, as stated by Lipka,[7] "Simple curves

Table III Temperature and Average Shear Stress for Removal of Ice from Various Surfaces

Surface Material	Temp F	Average Shear Stress, psi
Anodized Aluminum	1	356.7
	2	323.2
	3	321.4
	4	303.5
Stainless Steel 1018	2	248.3
	3	241.3
	4	237.9
	5	235.8
	6	231.3
White Teflon	1	75.1
	2	68.2
	4	66.3
Black Teflon	1	92.4
	2	45.0
	3	55.0
	4	49.4

Table IV Shear Stress Temperature Data For Removal of Ice From Anodized Aluminum

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
342.2	1	356.7	14.5	14.4
351.8	1		4.9	
356.6	1		0.1	
376.3	1		19.6	
270.8	2	323.2	52.4	36.2
290.8	2		32.4	
330.4	2		7.2	
336.4	2		13.2	
339.8	2		16.6	
370.7	2		47.5	
294.8	3	321.4	26.6	23.6
317.6	3		3.8	
320.8	3		0.6	
352.8	3		30.9	
265.3	4	303.5	38.2	28.6
300.2	4		3.3	
316.5	4		13.0	
332.1	4		28.6	

which approximate a large number of empirical data are the parabolic and the hyperbolic curves. The equation of such a curve is $y = a x^b$, parabolic for b positive and hyperbolic for b negative."

Using the method of least squares the values of a and b in the equation $S = a T^b$ were calculated. S was the arithmetical average of the shear stress in psi for a given surface at a given temperature, T, in degrees Rankine.

Table I was prepared to indicate how closely the calculated curve fit the experimental data. The tabulation shows the stress as calculated from the equation for each temperature, the difference between the experimental and the curve value, and the percent variation from the experimental. Fig. 2 is a plot of stress versus temperature in degrees Fahrenheit. The curve is the derived equation and the spotters represent the arithmetical averages of the experimental data.

When the absolute temperature was used, the equations involved constants of quite large magnitude so that to obtain more meaningful results further manipulation was used. The Rankine temperature, T, was replaced here by

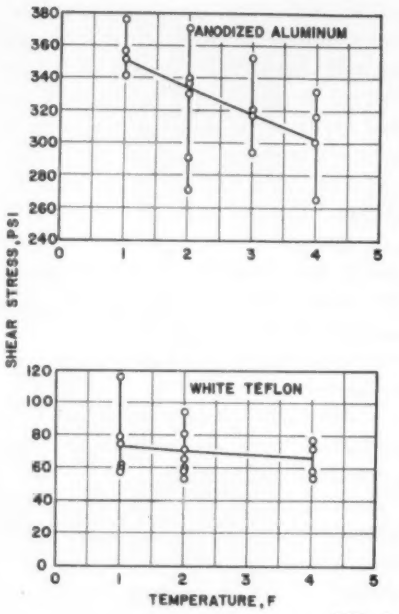

FIG. 3

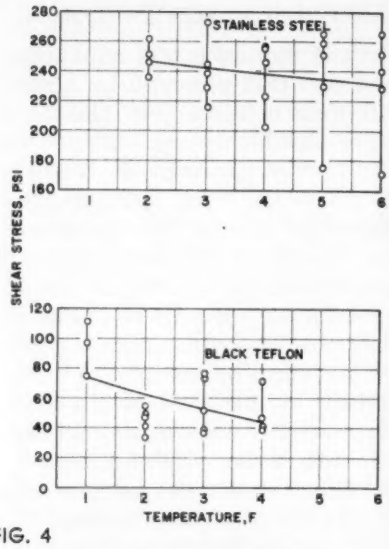

FIG. 4

Figs. 3 and 4 show the shear stress versus temperature for individual tests. Figs. were plotted with curve that fits the desired equation

460 + t, where t is in degrees Fahrenheit. Then the equation $S = a(460 + t)^b$ was expanded by the binomial theorem. In the resulting expansion only those terms were recorded which were needed to keep the shear stress value within plus or minus 0.5 psi of the value calculated with the original equation. The resulting equation is of the form, $S = c(1 - dt + et^2 - ft^3)$. Table II is a tabulation of the values of the constants for each surface which satisfy the above conditions.

DISCUSSION OF DATA AND RESULTS

Table III gives for the four surfaces tested the arithmetical averages at several temperatures in the temperature range, 1-6 F. The results shown in Table III and in Fig. 2 indicate a definite tendency toward decreasing strength with increasing temperature and large differences in adhesive strength for the various types of surfaces. The ice adheres most strongly to the anodized aluminum surface, next to the stainless steel, and then much more weakly to the two Teflons, white and black.

Ice will adhere more strongly to an oxidized metal surface; the anodized surface being a completely oxidized surface verifies this statement. According to results mentioned in the literature survey, the data are thus in correct order. The stress for metallic surfaces is greater than for organic surfaces and the stress for the completely oxidized surface is greater than that for the incompletely oxidized surface.

In Figs. 3 and 4 the shear stress values for individual tests are plotted along with the curve that fits the derived equation. These same values are tabulated in Tables IV, V, VI and VII. The values have been grouped numerically from

Table V Shear Stress Temperature Data for Removal of Ice from Stainless Steel

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
235.5	2	248.3	12.8	9.4
246.0	2		2.3	
248.6	2		0.3	
249.6	2		1.3	
261.8	2		13.5	
216.2	3	241.2	25.0	19.0
228.8	3		12.4	
239.3	3		1.9	
244.9	3		3.7	
245.6	3		4.4	
272.6	3		31.4	
202.5	4	237.9	35.4	21.5
222.9	4		15.0	
224.4	4		13.5	
246.2	4		8.3	
255.4	4		17.5	
257.1	4		19.2	
257.4	4		19.5	
174.6	5	235.8	61.2	36.8
229.6	5		6.2	
250.5	5		14.7	
258.8	5		23.0	
265.7	5		29.9	
170.8	6	231.3	60.5	36.4
228.6	6		2.7	
240.0	6		8.7	
251.4	6		20.1	
265.5	6		34.2	

the lowest to the highest for each temperature. These tables have been included as the statistical analysis of the data. The arithmetical mean has been tabulated as, "the arithmetic mean of a sample is the best estimate of the mean of a population."[9] The standard deviation is included as, "the standard deviation of a sample is the best estimate of the scatter in the population and is used to represent the spread."[9] The same author also claims that a sample usually has two-thirds of the data within the spread, arithmetic mean plus or minus the standard deviation. In the fifteen groups of samples this latter statement can be seen to be true.

From Table I, it can be seen that the deviations are small except for the black Teflon. This result was expected as the black Teflon surfaces used were not uniformly alike in roughness as were the other surfaces. Thus, these results have an additional variable involved.

Numerical accuracy of the stress values is high. The one measurement involved was the gross load of the load pan and lead shot at the instant of separation of the interface. The scale used was checked for accuracy with dead weights and was found to be accurate to the ounce. At the same time the sensitivity was determined and even at the loads involved the scale was found to be sensitive to weights of less than one ounce. These loads were multiplied by three, the lever arm ratio, which was correct as measured in several trials made with an engraved steel scale. The ounces were converted to the nearest one-tenth of a pound. The average values then should be correct to within two-tenths of a pound.

Table VI Shear Stress Temperature Data for Removal of Ice from White Teflon

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
57.6	1	75.1	17.5	24.2
60.0	1		15.1	
61.9	1		13.2	
75.0	1		0.1	
78.2	1		3.1	
117.8	1		42.7	
53.1	2	68.2	15.1	13.6
58.4	2		9.8	
59.3	2		8.9	
61.7	2		6.5	
66.0	2		2.2	
72.2	2		4.0	
81.0	2		12.8	
93.8	2		25.6	
54.4	4	66.3	11.9	10.9
56.4	4		9.9	
58.9	4		7.4	
72.6	4		6.3	
77.1	4		10.8	
78.2	4		11.9	

Table VII Shear Stress Temperature Data for Removal of Ice from Black Teflon

Shear Stress psi	Temp. F	Arith. Avg. psi	Absolute Value Stress—Avg. psi	Standard Deviation psi
74.8	1	92.4	17.6	15.8
96.9	1		4.5	
105.4	1		13.0	
32.8	2	45.0	12.2	8.3
40.9	2		4.1	
47.1	2		2.1	
48.8	2		3.8	
54.4	2		9.4	
36.0	3	55.0	19.0	18.9
38.4	3		16.6	
51.6	3		3.4	
72.6	3		17.6	
76.5	3		21.5	
38.4	4	49.4	11.0	15.3
41.3	4		8.1	
46.1	4		3.3	
71.8	4		22.4	

CONCLUSIONS

Work carried out in this experiment, including the preliminary phases, confirms that ice is not a uniform structure of crystals that has a unique strength of adhesion. Rather, ice is formed with imperfections between crystals that make weak spots, which influence the results obtained from strength tests. However, on the average there is a strength of adhesion which is a characteristic of the material to which the ice is frozen and which is a function of the temperature. It is believed that tests carried out in the same manner will provide results within the same limits determined in this experiment. It is also believed that the equations derived may be extrapolated beyond the range involved. According to observations in this experiment, if carried above 10 F, slow sliding will be present before the sudden failure. The prediction for the failure strength of thicker samples of ice is also open to question, as the larger volumes provide greater possibility of flaws in the ice which will produce different results.

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UNITED ENGINEERING CENTER NOVEMBER 9;

J. H. CANSDALE
Assistant Secretary

"This is a day of rejoicing among the 300,000 engineers in the United States. We have dreamed of this building for many years. We have believed that by housing our twenty different societies under one roof we would have more unity of purpose in our professions. We have believed that we could be of more service to the American people."

With these opening remarks, former President Herbert Hoover officially dedicated the United Engineering Center, the headquarters of 20 engineering societies, including ASHRAE.

Mr. Hoover spoke before about 700 persons in the Center's auditorium. Closed-circuit television carried his message to an over-flow audience in the Exhibition Hall. The occasion was an historic and long-awaited one for engineers. Mr. Hoover stated:

"Our engineering professions have two major responsibilities to the American people.

"It is our duty to take the discoveries in science from all over the world and translate them into new inventions, new processes and new facilities which will keep the American standard of living ahead of all the world. We have long since proved that our nation has the creative genius to do it. The exchange of ideas facilitated by this Engineering Center will give even more assurance of this service.

"We also have a responsibility to see that we maintain the supply of trained scientists and engineers needed by our growing economy and our national defense. I regret to say that we have fallen behind the Communists in this field. But with this new united front we can provide the remedy."

Among the 60 distinguished guests, United Engineering Trustees and Society Presidents on the platform with Mr. Hoover were Cardinal Spellman; Rev. Dan Potter, Executive Director of the Protestant Council of Churches; Rabbi Arthur T. Buch of the Congregation Shaare Zedek; Mayor Robert F. Wagner; representatives from Governor Rockefeller and the Commissioner of Com-

Former President of the United States, Herbert Hoover, speaks from the UEC auditorium podium

DEDICATED

John Everetts, Jr., Attends Ceremonies

Raising of the flag was a prelude to the dedication ceremonies

merce of New York; and John Everetts, Jr., President of ASHRAE. Other ASHRAE representatives, seated in the audience, were Merrill F. Blankin, Chaiman of the ASHRAE-UEC Fund Raising Campaign; Robert C. Cross, Executive Secretary; and members of the headquarters staff.

Earlier, Center personnel had watched as the flag was raised officially in front of the building and a proclamation was read from Mayor Wagner proclaiming Engineers' Week from Nov. 5-11 and naming 47th Street Engineers' Plaza, during this period. Music was provided by an Air Force band.

Willis F. Thompson, President of the United Engineering Trustees, was master of ceremonies during the hour and a half affair in the auditorium. Highlights were invocations by the three church dignitaries; a short greeting by Mayor Wagner and the presentation of the Key to the City; the reading of messages from President Kennedy, Governor Rockefeller and the heads of state of a number of foreign countries; and the presentation of the 1961 Hoover Medal to Dr. Mervin J. Kelly, former Chairman of the Bell Telephone Laboratories. The award, first given to Mr. Hoover in 1930, is sponsored jointly by several national engineering societies.

Representing the new generation of engineers, "who had little or nothing to do with the creation of this structure but who will surely benefit most from its existence," was William Hallerberg, 21-year-old honor senior majoring in metallurgy at the Missouri School of Mines and Metallurgy in Rolla.

Mr. Hallerberg commented, "The engineers of my day will surely produce results that will surpass anything that we now have. Not because we will be more intelligent, not because we will be better as individuals, but because our predecessors have already placed many stepping stones into the stream of knowledge that will provide the beginning path which may then be extended into the now unknown."

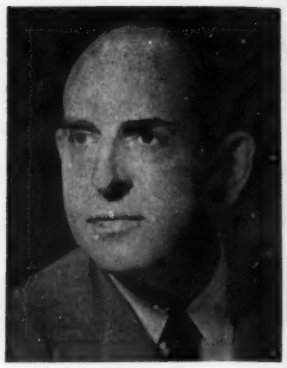
Principal speaker was Dr. Eric A. Walker, President of Pennsylvania State University. In his address Dr. Walker emphasized that engineers and engineering have shaped the American economy into what it is today, that their work is critical to our national defense and to the survival of democracy and the ideal of the dignity of the individual.

He stated, "To an engineer, knowledge is not an end to itself but is simply the raw material from which he fashions structures, machines and processes."

Among those on the speakers' platform: Peter Paul Miller, Director of the City of New York office of the New York State Dept. of Commerce, representing Governor Rockefeller; H. M. Lewis, President of the American Institute of Consulting Engineers; John Everetts, Jr., President of ASHRAE; E. R. Needles, Engineers Joint Council; the Reverend D. Potter

PRESIDENT'S PAGE
JOHN E. DUBE
Guest Sponsor

"... the end of deficit spending..."



JOHN E. DUBE
Treasurer
ASHRAE

The 1960-61 fiscal year which ended June 30, 1961, saw the end of deficit spending for our Society. We achieved a small operating surplus and after allowances for admission fees and investment income, our members' equity rose by about $22,000. This figure does not reflect a further increase of net worth of $81,000 due to one-half of the last exposition income which was received during the last fiscal year, but is budgeted to be spent for research grants this year when there will be no exposition.

The Board of Directors squarely faced its responsibility and made the important – and in some instances – unpopular decisions which made it possible to eliminate red ink while sacrificing none of the real goals of the Society.

Now, if we were able to balance last year's budget, why was it necessary to raise this year's dues by 10%? That is a fair question and the answer can be found in the fact that this year we have to meet substantial additional obligations. These are NEW obligations, there is no carry-over from the past – nothing has been swept under the rug. For example, under the new set-up, we will meet expenses for Research Administration out of General Society Funds, and thus, we will actually spend more dues money on research than we did last year. Our JOURNAL editor is within two years of retirement and in the interest of good management, the Board authorized the employment of an associate editor.

This year we have moving expenses; we expect to spend substantially more for stationery and office supplies which, in view of the move and change of address, were permitted to run low last year. Our rent, although reasonable, will be higher. To aid the work of the Membership Committee, we budgeted more money for a membership campaign and public relations. And we also will publish a new and better membership roster.

We also broadened the scope of our staff retirement program . . . to include the few employees who could not participate originally due to their advanced age. The Society was committed to meet their pension payments out of future operating funds . . . a practice which could have developed into a substantial burden. We also budgeted for travel expenses of four additional committees . . . a rather related implementation of a decision made by the Board of Directors during the Dallas meeting.

These new and necessary items represent an increase of annual expenditures in excess of $60,000, which is approximately $20,000 more than the additional income resulting from the increase in dues. Consequently, this year's budget had to be balanced by projecting further increases in the operating efficiency of the Society despite the fact that last year we experienced a $55,000 reduction in headquarters' salaries (this figure does not include Cleveland personnel).

This year we started a new budgeting practice, namely, the allocation of all headquarters' expense to the following six major accounts:

General & Administrative
Membership
Finance
Technical
Research Administration
Publications (GUIDE and JOURNAL)

With the aid of this allocation we determine how our dues dollar is budgeted to be spent this year. The tabulation is based on 17,600 dues-paying members and average dues of $25.

BREAKDOWN OF DUES INCOME 1961-62 Budget	Per Dues Paying Member
1. Society Headquarters	
(a) General & Administrative	$ 5.76
(b) Membership	3.04
(c) Finance	1.78
(d) Technical	1.33
2. Research Administration	1.82*
3. Net Cost of GUIDE	1.49
4. Net Cost of JOURNAL	2.34
5. Transactions	.31
6. Membership Roll	.57**
7. Travel Expenses – Officers, Directors & Committees	1.62
8. Chapter Allowances & Supplies	2.16
9. National Meetings	1.18
10. Membership Campaign	.12
11. Certificates, Medals & Awards	.14
12. ASA, EJC, IIR, UEC	.69
13. Moving Expense	.46***
14. Surplus	.19
	$25.00

* Research Administration only—exclusive of $120,500 for research promotion and research grants financed from Exposition income and industry contributions.
** This expense will be recovered partially in the following year.
*** Non-recurring except $3,500 for leasehold improvements in each of the next two years.

When the Board of Directors approved the increase in dues from $25 to $27.50, it also approved the elimination of chapter allowances as of July 1, 1962, subject to concurrence by the Regions Central

(*Continued on page* 90)

Integrated *Radiation-Convection Cooling System*

design and performance

Committees of the Illuminating Engineering Society recently have prepared new tables of recommended footcandle levels for a variety of visual tasks. These values are two to three times those recommended formerly. This use of higher lighting levels is altering the entire economics of building cooling systems. In attempting to utilize lighting levels of 100 through 250 footcandles, we found that the increased costs associated with "all air" conditioning systems required by the higher lighting levels often prohibited their use.

Studies indicate that in an office building provided with 100 fc of general illumination, the lighting load accounts for approximately 42% of the total air conditioning requirements. If the fc level is increased to 400 fc, the lighting heat load accounts for approximately 72% of the total load. Higher concentrations of electrical energy are inevitable in modern buildings – not only for lighting, but for business machines and other equipment.

Instantaneous heat gains can be controlled more effectively by building cooling systems which respond and take into account the inter-relationship that exists between outdoor environment with its periodic changes, and indoor environment with its various sources of sensible and latent heat load and the building structure with its capacity for heat storage. To accomplish this, the character of the building interior and outdoor thermal load sources must be analyzed in relation to the available heat transfer mechanisms of the building cooling system. A significant

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Associate, ASHRAE

portion of the solar heat gains as well as lighting and equipment load are transferred by radiation. The human body also gives off heat by radiation, convection and evaporation of moisture.

For optimum conditions, heat removal should be accomplished by both radiation and convection. Furthermore, it has been demonstrated that comfort and system efficiency are improved when space thermal gains are removed in proportion to and by a mechanism identical to that releasing it. This paper presents a system staging radiant heat transfer, convection heat transfer and latent heat removal and designed to achieve these optimum conditions and couple their effects with building heat storage.

DESIGN PRINCIPLES

The system is so arranged that various heat gains to the space are removed at the highest possible temperature levels. This results in greatly reducing the installed refrigeration requirements. This system and the optimum arrangement of equipment permits handling the air conditioning load with higher water temperatures and greatly reduced air quantities. This results in smaller air conditioning apparatus and auxiliary equipment.

The design of air conditioning systems usually is based on energy balances and efficiencies that result from application of the principle of conservation of energy that follows from the First Law of Thermodynamics. The Second Law of Thermodynamics, however, has extreme significance in the design of building cooling systems since the character of these systems is to remove heat at one temperature level and reject it at some higher temperature level. Systems design calculations using the Second Law of Thermodynamics as a basis for evaluation indicate the relative importance of each item of the cooling load and the optimum efficiency which is possible by the correct choice of the system arrangement and operating conditions.

The Second Law of Thermodynamics tells us that all irreversible spontaneous processes are accompanied by a degradation of energy. An increase in entropy accompanying the addition of a given amount of heat load to a building cooling system is caused by lowering the temperature at which this heat is removed. In applying the Second Law of Thermodynamics to the design and arrangement of building cooling systems, the temperature exchange level at which heat is removed must be staged according to the load source. In order to utilize this principle, dynamic integration of the mechanical and electrical building systems is necessary. No longer can these systems be considered independently of each other. The selection and design of each system must be developed simultaneously. Dynamic integration is brought

Gershon Meckler is Director, Meckler Engineering Company. This paper was prepared for presentation at the ASHRAE Semiannual Meeting, St. Louis, Mo., January 29-February 1, 1962.

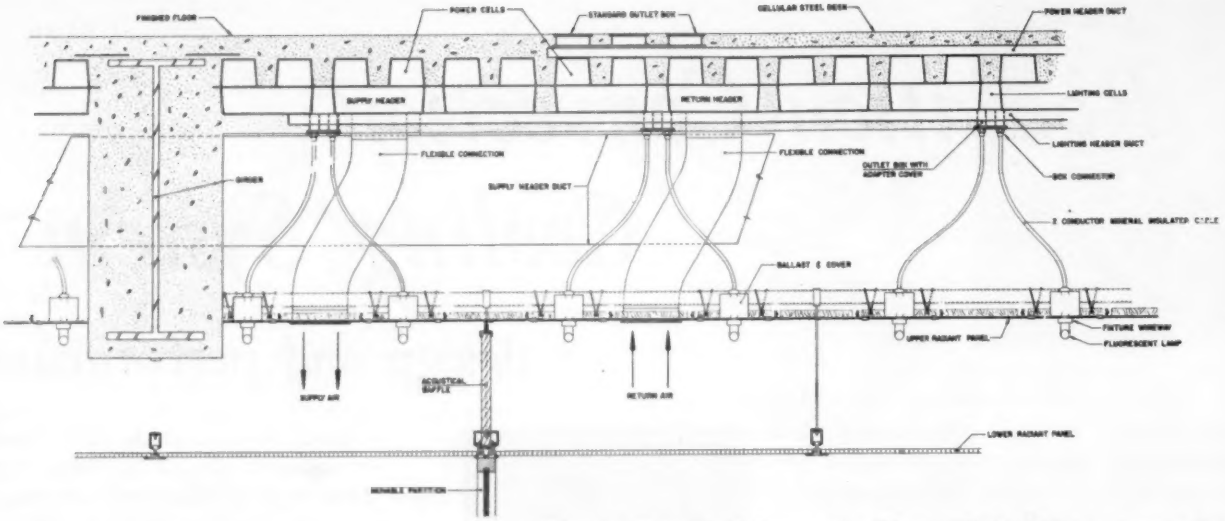


Fig. 1 Integrated lowered ceiling, cellular floor arrangement

about through the design of dual purpose products which are common to two or more building systems permitting the direct interaction of system energies.

RADIATION-CONVECTION

The most important function of the system is to provide flexible temperature and humidity control of a multiplicity of occupancy zones utilizing a constant temperature air distribution system and integrated ceiling and wall radiant panels which respond to all space heating or cooling loads. Ventilation, space humidity control and comfortable air motion is maintained with supply air rates of approximately 0.6 cfm per sq ft of conditioned space. When properly designed any desired temperature or humidity range can be maintained in any zone independent of sensible or latent load variations in all other zones.

The integrated radiation-convection cooling system is a split system which permits the integration of lighting components, such as a water cooled radiant lighting troffer or radiant luminous ceiling, with other building systems to improve the luminous efficiency of a lighting system and to remove directly at its source a substantial portion of the associated thermal load. Only minimum ventilation is supplied when water tubing is used in place of bulky air ducts.

Referring to Fig. 1, in the case of the radiant luminous ceiling system the integrated upper panel is so arranged that it removes heat generated directly by the ballast and contributes to the cooling of the ceiling cavity.[1] The lower radiant louver is arranged to diffuse the light and provide radiant cooling for the occupied space below. The lower ceiling section of the radiant luminous ceiling consists of an arrangement of water cooled supporting tees and a heat transmitting light diffusing louver. The tees are arranged on a modular basis to provide a maximum flexibility for future space changes as well as temperature control changes.

Referring to Fig. 2, the water cooled luminaire consists of a specially designed luminaire assembly in which the upper housing and the radiant louver located as a light diffusing grid below the lamps are cooled by temperature staging water circuits. The upper housing corresponds in function to the upper radiant acoustical panel and luminaire assembly at the radiant luminous ceiling. The light diffusing louver corresponds in function to the lower louver section of the radiant luminous ceiling. The use of either type of ceiling system will depend upon building and occupancy requirements. However, the control of the space or operation of the basic mechanical system remains the same. Whether a radiant luminous ceiling or individually water-cooled luminaires are utilized, each individual space module can have its own temperature control.

The radiation-convection cooling system is designed to provide maximum flexibility and to discriminate between the type of load occurring at the exposed perimeter and at the interior of the building. Fig. 3 illustrates the use of the radiation barrier for a 100% floor to ceiling glass exterior curtain wall. The solar energy enters the glass curtain wall on the venetian blind surface except for an amount that enters the space directly. At the perimeter of the building, where more than 75% of floor to ceiling exterior curtain wall consists of glass, air is introduced at the interior of each space and is directed toward the curtain wall. The perimeter return air grilles are located on the ceiling and on the floor between the exterior curtain wall and the venetian blind.

It is possible to maintain a higher surface temperature on the inside surface of the glass curtain wall and use return air to remove some heat by conduction-convection. The venetian blind acts as an intermediate surface between the glass curtain wall and the perimeter radiation barrier panels. The perimeter panels view the entire blind surface and exchange heat directly by radiation from the venetian blind. The radiation barrier panels are controlled on the basis of solar heat gains and the outside air temperature. For buildings with less than 75% floor to ceiling glass in exterior wall, the radiant wall panel can be recessed. Both the perimeter return air and interior return air are connected to the cellular steel floor which returns the air to a common return air header which is connected to the vertical return air shafts at the building core.

OPERATION OF INTEGRATED RADIATION-CONVECTION SYSTEM

Air Systems – Fresh dehumidified air is introduced to the occupied spaces to provide ventilation and to remove latent heat. A minimum of outside air is filtered and dehumidified by means of chemical air conditioners which bring air into contact with hygroscopic solution. This solution removes or adds moisture in selected amounts. All required dehumidification for removal of space latent heat is provided by the chemical air conditioners. Outside air introduced for ventilation purposes is dehumidified to a lower than conventional absolute humidity level and then is mixed with recirculated air from the space to provide proper space latent heat control. The total mixed air then is cooled sensibly to the desired unzoned discharge air temperature at the proper absolute humidity level.

One of three seasonal "occupied" operating air cycles commences, depending upon the prevailing outside air conditions and indoor humidity control requirements.

Winter Cycle – With the outside air humidity below the winter supply air design humidity and the outside air temperature below the winter supply air design temperature, all outside air is preheated as required. Heat for humidification in the chemical air conditioner is provided by hot water preheat coils which exchange condenser water with entering outside ventilation air.

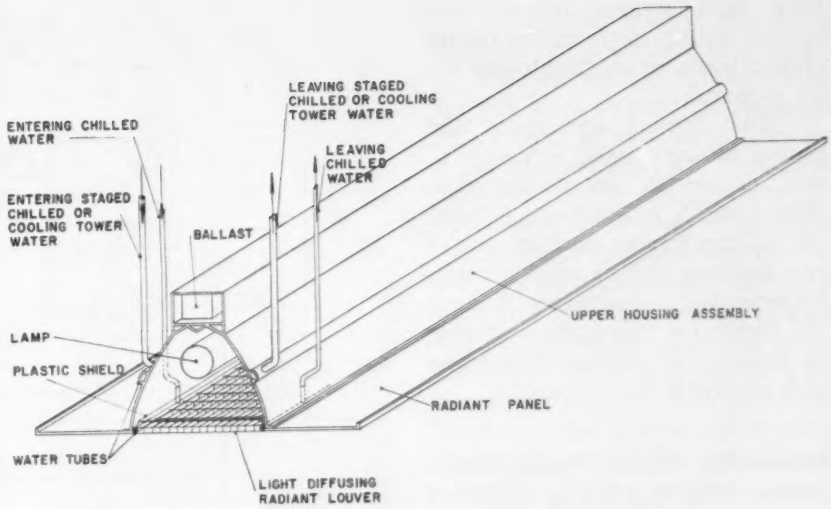
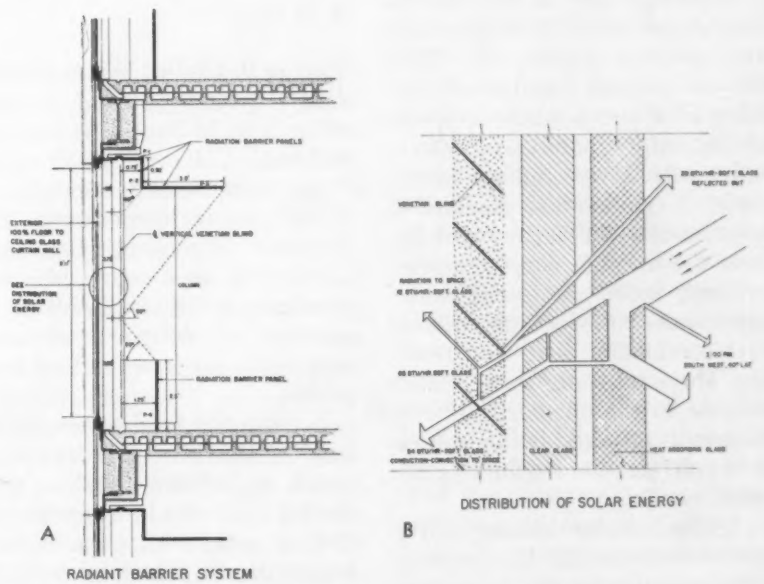


Fig. 2 Isometric view of water cooled luminaire



Figs. 3A and 3B A–Radiant barrier system. B–Distribution of solar energy

Intermediate Cycle – With the outside air humidity above the winter supply air design humidity, but below the summer supply air design temperature and humidity, the mixing dampers on all air handling units modulate to maintain the winter supply air design temperature. The chemical air conditioners are inoperative during intermediate summer cycle.

Summer Cycle – With the outside air humidity above the summer supply air design humidity or outside air temperature above the summer supply air design temperature, tower water or chilled water is circulated through chemical air conditioner coils to remove the latent load as required. The mixed return and dehumidified outside air is circulated to all air handling units, which cool this air to the summer supply air design temperature by means of chilled water. Dilute solution from the chemical air conditioner is reactivated in the chemical regenerator which utilizes steam to concentrate the hygroscopic solution.

RADIATION SYSTEMS

Integrated radiation-convection systems permit increasing efficiency by removing sensible load at successively higher temperature levels by means of temperature staging. Temperature staging can be brought about by series arrangement of building water circuits or arrangement of two or more separate water circuits operating at different temperature levels. Radiation barrier panels operate on cooling or heating cycle depending upon condition of the sun at each orientation or outside air temperature. The lower radiant louver panels operate in conjunction with the constant temperature air system to supply cooling or heating as determined by individual zone space thermostats responding to variations in space air temperature. The upper radiant luminous ceiling assembly or water-cooled luminaire operate to remove heat

from light sources below year-round using high temperature chilled water or cooling tower water as required.

Two typical equipment and piping arrangements, Scheme "A" and Scheme "B," are illustrated in Figs. 4 and 5, respectively. Scheme "A" utilizes temperature staging of two separate chilled water circuits by series flow arrangement. Scheme "B" illustrates temperature staging by means of separate chilled water and cooling tower circuits.

Scheme A: Chilled Water Distribution—Referring to Fig. 4, chilled water from refrigeration machine (1) is circulated first to all CHC coils in all air handling units, taking approximately an 8 F rise in water temperature. From the total water quantity leaving the CHC coils, an amount capable of producing a 5 F rise in water temperature at design conditions is circulated to the lower radiant louver panels. A temperature gradient of approximately 5 F is required between the average panel temperature and mean circulated water temperature to overcome radiant panel conduction heat flow resistance. The total water then is made available to the chemical air conditioners as required before returning to refrigeration machine (1) for further cooling.

Chilled water leaving refrigeration machine (2) is circulated first to radiation barrier panels covering each exterior wall. Temperature gradient of 2 F is required to overcome panel conduction heat flow resistance. The total water leaving the radiation barrier system is circulated next to the upper radiant luminous ceiling assembly or luminaire housing rising approximately 10 F under design load. A temperature gradient of 3 F is required to overcome panel conduction heat flow resistance. The total water is returned to refrigeration machine (2) for further cooling.

Scheme A: Cooling Tower Water Distribution—Cooling tower water is circulated to all refrigeration condensers and chemical air conditioners as illustrated. The temperature rise of total water circulated will depend upon specific job conditions, but should be in the range of 10 to 12 F.

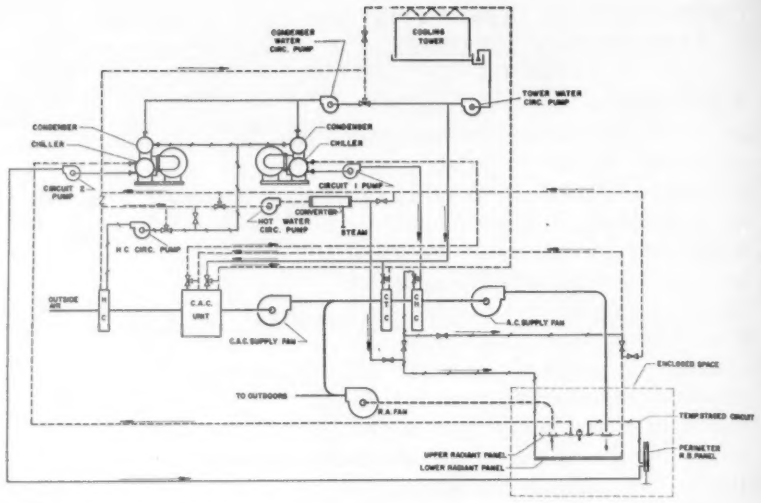


Fig. 4 Scheme A

Scheme B: Chilled Water Distribution — Referring to Fig. 5, chilled water from chiller (2) is circulated first to all CHC coils in all air handling units, taking approximately an 8 F rise in temperature. From the total water quantity leaving the CHC coils, an amount capable of producing a 5 F rise in water temperature at design conditions is supplied to the lower radiant louver panels.

From the total water quantity leaving the lower radiant louver panel, an amount capable of producing a 5 F rise in water temperature is supplied to the radiation barrier panels. The total water then is made available to the chemical air conditioners as required before returning to chiller (2) for further cooling.

Scheme B: Cooling Tower Water Distribution—Cooling tower water is circulated to all refrigeration condensers and air conditioners as illustrated. During winter operation, chiller (1) is operated to remove heat from cooling tower water circulated to upper radiant luminous ceiling assembly or upper luminaire housing. Heat is extracted as required from the 80 to 85 F cooling tower water and used to supplement the building heating load through the condenser water circuit. During summer operation, chiller (1) is inoperative and 80 to 85 F water from the outdoors is available through the use of the cooling tower directly.

Condenser Water Distribution – Under both Scheme "A" and Scheme "B," condenser water leaves the condenser of each refrigeration machine at 125 F and a portion of it is circulated first to the outside air coil. The condenser water leaves the outside air coil and mixes with the remainder of the 125 F condenser water before entering the perimeter heating system loop. A quantity of water equal in amount to that entering the perimeter heating system loop is returned to the condenser water system, completing the circuit. A portion of the condenser water continuously is exchanged with an outdoor cooling tower or air cooler in relation to operating refrigeration demands.

For each 1,000 MBH required for heating, approximately 80 kw of electricity must be consumed by the refrigeration machine operating between 65 F evaporator temperature and 125 F condenser temperature. Consider that electricity is purchased at $0.013 per kw-hr and that 5 lb steam for heating can be purchased at a cost of $1.20/1,000 lb. Therefore, the steam cost associated with supplying 1,000 MBH for heating equals $1.25. Provided sufficient heat energy is available at the refrigeration condenser, the cost of electricity necessary to add 1,000 MBH to the outside air and simultaneously remove approximately 68 ton of interior load by operating the existing refrigeration equipment as a heat pump is only $1.04.

Thus, the cost of utilizing heat generated by means of the condenser heat pump circuit repre-

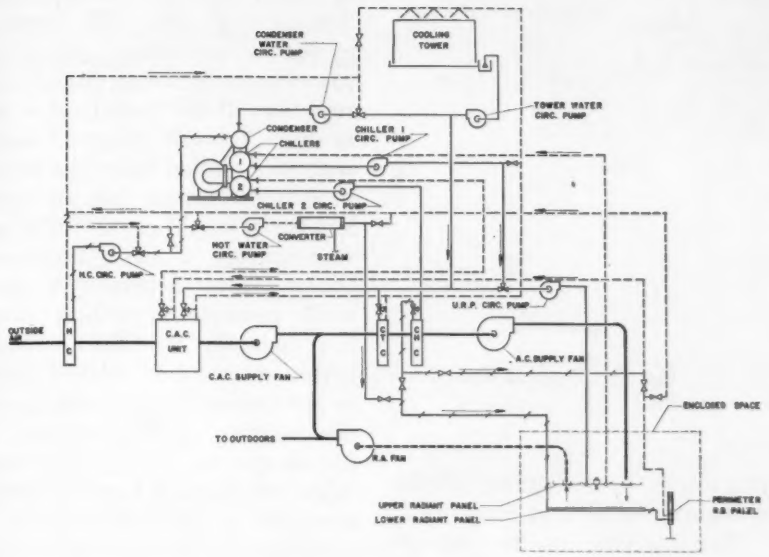


Fig. 5 Scheme B

sents approximately an 18% savings over the cost of steam for heating with no cost associated with the removal of space load. All this is accomplished by operating the refrigeration machine as a heat pump utilizing the building lighting system as a stable heat source.[2]

SPACE HUMIDITY CONTROL

The use of the cooling coil as a means of striking a satisfactory balance between sensible heat and latent load often imposes an unnecessary refrigeration load penalty. The air leaving wet bulb temperature is usually within one or two degrees of the desired dry bulb temperature. When relatively low supply air temperatures are required due to high thermal space loads, an excessive moisture load is imposed upon the refrigeration system. Unnecessary low space humidities result from this situation.

Koch, Jennings and Humphreys presented the results of their most recent attempt of research designed to correlate human comfort with space environment.[3] In this study their measurements were restricted to the effects of air dry bulb temperature and humidity. In these tests the subjects wore normal summer clothing and were seated. Air motion in the test room was at a minimum and all room surfaces were maintained at approximately the dry bulb temperature. Tests were conducted at dry bulb temperatures ranging from 68-94 F with relative humidities from 20 to 90%. The paper illustrated that the line of optimum comfort is a straight line in a dry and wet bulb temperature plane and that the effect of relative humidity change is small. In short, the line of optimum comfort for both winter and summer is approximately a straight line range from 77.6 F dry bulb at 30% RH to 76.5 F at 85% RH. According to the authors, the line of optimum comfort is "only slightly dependent upon humidity."

The results of a similar study were published by Warden, Fahnestock and Galbraith.[4] The authors showed that a comfort vote closely corresponding to criteria used by Koch, Jennings and Humphreys in their paper was obtained along a comfort line established by the points 76 F dry bulb temperature at 40% RH, and 75 F dry bulb temperature at 80% RH.

It appears that if the space temperature is controlled at about 75-76 F, the space relative humidity may vary within a rather broad range without producing any noticeable discomfort to occupants. In order for the integrated radiation-convection system to maintain the appropriate space relative humidities by independent temperature and humidity control during winter and summer occupied cycle, the proper outlet air humidity leaving the chemical air conditioner must first be determined in relation to the outside air requirements of each occupancy zone.

Refer to Fig. 6 for the procedure used to determine the required outlet air humidity leaving the chemical air conditioner for both winter and summer operation. Once the required design space latent load and the minimum ventilation air rate are established, the chemical air conditioner outlet air humidity for winter and summer design conditions is calculated by means of Equations (1) through (3). The required outlet air humidity value obtained from Equation (1) is inserted in Equation (2) or (3), from which the maximum space latent load permissible is calculated. If the anticipated maximum load is beyond that calculated using the minimum ventilation air rate, then the outside air quantity is increased until Equation (2) or (3) is satisfied, guaranteeing that the humidity in the conditioned space shall be below 60% relative humidity during summer operation.

Fig. 6 Outlet air humidity

THE DETERMINATION OF THE REQUIRED OUTLET AIR HUMIDITY LEAVING THE CHEMICAL AIR CONDITIONER IS ACCOMPLISHED BY MEANS OF THE TWO GENERALIZED RELATIONSHIPS GIVEN BELOW:

OPERATING CYCLE	SPACE CONDITION TEMPERATURE (°F)	% RELATIVE HUMIDITY DESIGN	% RELATIVE HUMIDITY MAXIMUM	EQUATION CONSTANTS K_1	K_2	K_3
SUMMER	75	50	60	44.2	65	78
WINTER	75	30	35	26.5	39	46

REFERRING TO THE TABLE ABOVE, THE REQUIRED OUTLET AIR HUMIDITY LEAVING THE CHEMICAL AIR CONDITIONER, SUMMER OR WINTER, IS GIVEN BY THE FOLLOWING EQUATION:

$$G_{os} = G_{ow} = \frac{K_1 C_{os} - I_d}{0.68\, C_a} \qquad (1)$$

THE MAXIMUM SPACE LATENT LOAD PERMISSABLE TO INSURE THAT NO CONDENSATION WILL OCCUR ON A COLD SURFACE AND THAT NO CHANGE WILL RESULT IN SPACE COMFORT IS GIVEN IN THE FOLLOWING EQUATIONS:

WINTER

$$I_{mw} = \frac{(K_2 - G_{ow})C_{oa} + C(K_3 - K_2)}{1.47} \qquad (2)$$

SUMMER

$$I_{ms} = \frac{(K_2 - G_{os})C_{oa} + C(K_3 - K_2)}{1.47} \qquad (3)$$

SEE APPENDIX A-1 FOR COMPLETE DERIVATION AND NOMENCLATURE.

LOAD SOURCE	HEAT RELEASE MECHANISM - BTU/HR-FT^2 RADIATION	CONVECTION	MOISTURE
OUTSIDE AIR(1)	--	6.5	10.8
TRANSMISSION	3.5	2.5	--
PEOPLE(2)	2.2	1.1	3.1
LIGHTS(3)	20.5	20.5	--
SOLAR(4)	2.9	2.1	--
EQUIPMENT	1.7	1.7	--
STORAGE	--	3.5	--
FANS	--	1.9	--
TOTALS	30.8	39.8	13.9

(1) 0.3 CFM/FT^2 VENTILATION AIR
(2) 1 PERSON/75 FT^2
(3) 150 FOOTCANDLE INSTALLATION
(4) INSIDE VENETIAN BLINDS

Fig. 7 Typical design load

For the purpose of illustration, the maximum winter relative humidity was established at 35%; however, it may vary somewhat depending upon the exterior wall construction. The same procedure is repeated for winter operation and the minimum outside air rate that satisfies ventilation and both the winter and summer humidity control requirements is used to establish the mixture of outside and return air necessary for any specified occupancy zone. This procedure is repeated for all other conditioned spaces. Once the minimum outside air rates are obtained for all occupancy zones, it is possible to arrange for the proper outside air and return mixture at each air conditioner serving any combination of occupancy zones to establish a common humidity level satisfactory to all.

ANALYSIS OF SPACE LOAD

The instantaneous load in typical air conditioned spaces is generated by some combination of sun, lighting, people, transmission through walls or roof and glass, ventilation air, appliances, business and other machinery, etc. A significant portion of sensible load from these load sources appears as radiant heat. The subject system is arranged to view the space load in such a way as to optimize the amount of heat transfer by radiation and shift as much of the convection load to the water circuits as possible. The breakdown of a typical building space load into its corresponding radiation, convection and moisture (latent) load components is presented in Fig. 7. Notice that approximately 37% of the space load is generated at its source by radiation, 47% by convection and 16% as moisture.

The increase in the entropy $(\Delta S)_{T_r}$ of the building cooling system associated with the transfer of load (Q_1, Q_2) from individual sources (lights, people, equipment, etc.) at any corresponding temperature exchange levels (T_1, T_2) can be related mathematically at any temperature level assigned for heat rejection by the following equations:

$$(\Delta S)\ T_r = -\frac{Q_r}{T_r} + \sum_{i=1}^{i=n} \frac{Q_i}{T_i} \qquad (4)$$

$$\text{Where } Q_r = \sum Q_1 + Q_2 + \ldots Q_n \qquad (5)$$

Referring to Equations (4) and (5), the increase in system entropy of a building cooling system can be reduced by increasing the temperature exchange levels (T_1, T_2) at which load is removed from individual load sources (1, 2). Notice that as $T_1 \rightarrow T_r$ $(\Delta S)_{T_r} \rightarrow 0$.

Fig. 8 illustrates how the integrated radiation-convection system responds as a cooling system to the load in Fig. 7 under Scheme "A" and Scheme "B," respectively. Note the relationship of load removed by air as indicated from the psychrometric analysis and the corresponding load shifted to the water distribution circuits under each scheme. Although the psychrometrics remain the same under Scheme "A" and Scheme "B," almost all of the heat generated by radiation is captured by the water cooled circuits, in addition to a significant amount of load that otherwise has to be removed by air convection.

Referring to Fig. 8 the increase in system entropy corresponding to Scheme "A" and "B" operating cycles is calculated using 85 F tower water as the medium for heat rejection. If the same load as given in Fig. 7 were removed instead with 45 F chilled water the increase in system entropy (on an equivalent basis) would equal 0.0123 units, representing a 128% increase in entropy over Scheme "A" and a 200% increase in system entropy over Scheme "B." The increase in system entropy is related directly to the thermodynamic efficiency of the building cooling system. The use of systems energy is optimized when the Second Law of Thermodynamics is used to evaluate and compare the thermal efficiency of building cooling systems.

EFFECTS OF HIGHER LIGHTING LEVELS ON AIR CONDITIONING DESIGN

For lighting levels up to 75 fc the "all air" conditioning system has been satisfactory for removal of sensible loads associated with lighting and interior building occupancy. At 75 fc the amount of air necessary to remove the humidity load generated in interior office space is approximately the same as required to maintain proper space temperature. As the lighting level is increased beyond 75 fc (all other factors remaining the same), the air rate necessary to remove occupancy and lighting loads is increased beyond that necessary for humidity control to a point where it becomes uneconomical to use "all air" for space sensible cooling.

Beyond 75 fc, building floor-to-floor height in many cases must be increased beyond structural requirements to accommodate extensive ductwork between floors. The additional cost of increasing floor-to-floor height to provide air conditioning affects the material for stairs, elevator cables and rails, escalator equipment and vertical risers for plumbing, heating, air conditioning, wiring . . . in fact every mechanical, architectural and electrical building product that is installed between floors. Use of water cooled systems allows a reduction in building volume, material and distribution costs.

Consider the relative heat transfer ability of air and water to transfer space load from the en-

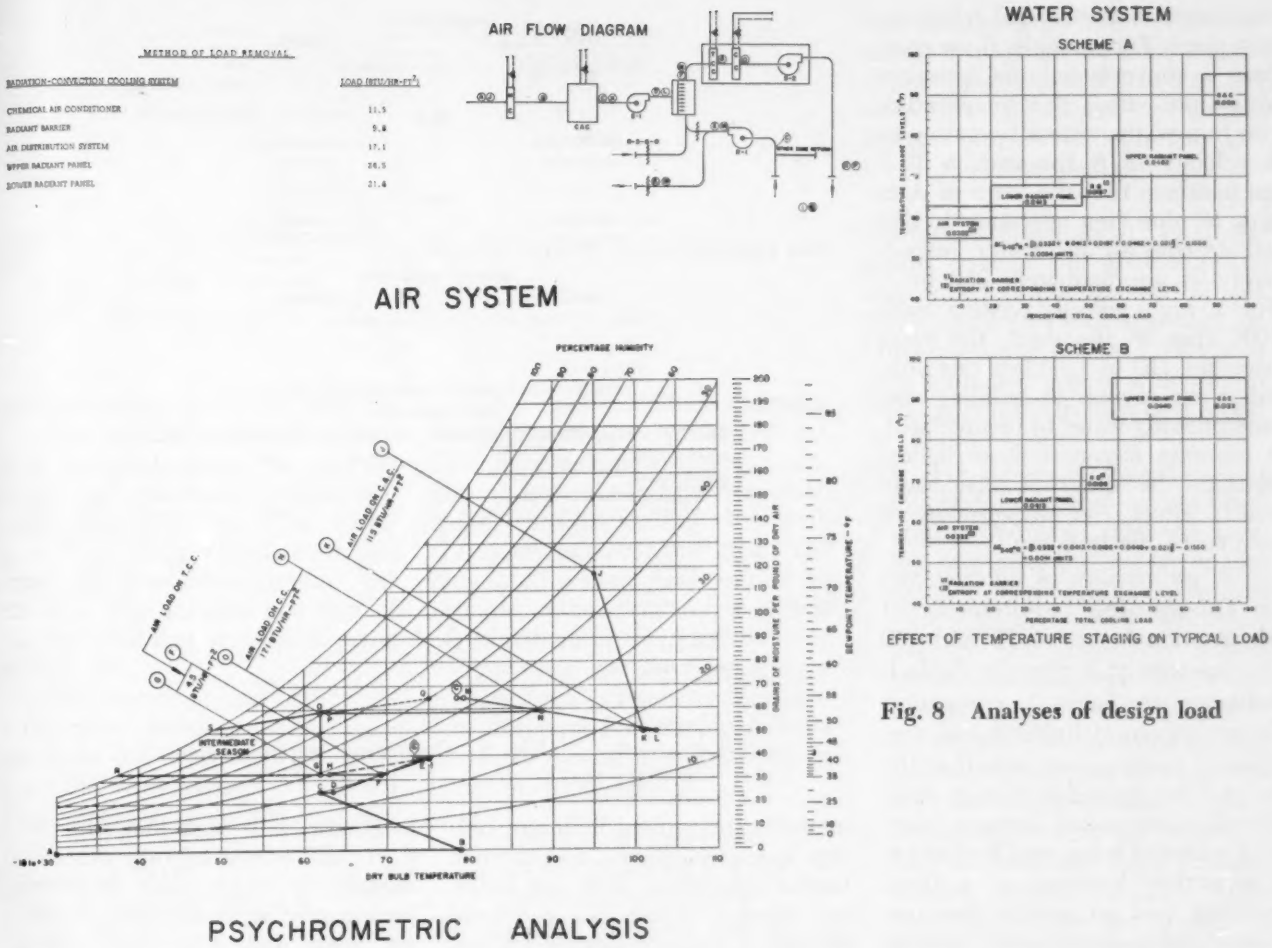


Fig. 8 Analyses of design load

TABLE I COST COMPARISON OF CONVENTIONAL AND INTEGRATED AIR CONDITIONING AND ELECTRICAL BULDING SYSTEMS

TBC[2]	LTG Cost[3] as % TBC	AC Cost[4] as % TBC	LTG Cost as % TBC	AC Cost as % TBC	Increase In LTG Cost as % TBC	Increase In AC Cost as % TBC	Increase In ARCH Cost as % TBC	Total Increase In BLDG. Cost as % TBC	% Increase in AC Operating Cost
75 FOOTCANDLE ILLUMINATION			150 FOOTCANDLE ILLUMINATION						
CONVENTIONAL LUMINAIRES[1]						CONVENTIONAL LUMINAIRES			
						2-F48T12/CW/H.O.			
18.00	13.4	19.8	20.7	31.0	7.3	11.2	0.4	18.9	51
20.00	12.1	17.9	18.7	27.9	6.6	10.0	0.4	17.0	51
25.00	9.7	14.3	15.0	22.3	5.3	8.0	0.4	13.7	51
						1-F48PG17/CW			
18.00	13.4	20.0	18.3	27.8	4.9	7.8	0.3	13.0	39
20.00	12.1	18.0	16.4	25.0	4.3	7.0	0.4	11.7	39
25.00	9.7	14.5	13.2	20.0	3.5	5.5	0.4	9.4	39
						INTEGRATED WATER COOLED LUMINAIRES			
						Savings in AC Cost as % TBC	Savings in ARCH cost as % TBC		% savings in AC operating cost
						2-F48T12/CW/H.O.			
18.00	13.4	19.8	23.9	16.1	10.5	3.7	2.0	4.8	33
20.00	12.1	17.9	21.5	14.5	9.4	3.4	2.1	3.9	33
25.00	9.7	14.3	17.2	11.6	7.5	2.7	2.3	2.5	33
						1-F48PG17/CW			
18.00	13.4	20.0	20.8	15.2	7.4	4.8	2.0	0.6	36
20.00	12.1	18.0	18.8	13.7	6.7	4.3	2.1	0.3	36
25.00	9.7	14.5	15.0	11.0	5.3	3.5	2.3	-0.5	36

(1) Lamp Combination in Luminaire is 2-F48T12/CW/S.L. for all comparisons at 75 footcandles.

(2) TBC denotes original total building cost in $/ft.² floor area at 75 footcandles.

(3) LTG denotes lighting system.

(4) AC denotes air conditioning system.

NOTE: Four (4) foot recessed luminaires were used in all comparisons.

vironment to the central refrigeration plant. For example, if we compare a conventional low pressure air system with a 20 F temperature rise to a water system operating at a velocity of 6 fps with a 10 F temperature rise, the ratio of duct area to pipe area necessary to distribute cold air and water, respectively, is approximately 350 to 1. For a high velocity system with 3000 fpm in the duct, the ratio becomes 210 to 1. When one considers the savings in building volume utilizing water to remove heat, it becomes apparent these figures indicate the order of magnitude which affect the non-productive volume of the building.

A comparison of lighting and air conditioning requirements at illumination levels of 75 and 150 fc, respectively, is given in Table I. Referring to Table I, comparing the architectural lighting and mechanical system costs, note that 150 fc can be provided (using integrated water-cooled fixtures operated with the Integrated Radiation-Convection System) at a total building cost no greater than required for conventional lighting and "all air" conditioning systems at 75 fc. For a building costing $20 per sq ft the 6.7% increase in total building costs associated with providing more lighting is offset by a corresponding 4.3% savings in air conditioning and a 2.1% savings in architectural cost. The same lamp source in a conventional luminaire at 150 fc, when used with an "all air" conditioning system, requires an 11.7% increase to the total building cost. Note that for a building costing $20 per sq ft the increase in lighting cost necessary to provide 150 fc amounts to but 4.3% of the total building cost. However, an additional 7.0% increase to the total building cost is associated with air conditioning required by this change. An operating savings of approximately 36% in the operation of the integrated air conditioning system can also be obtained at the 150-fc level. A 39% increase to the operating cost of the "all air" conditioning system would be experienced at the 150-fc level.

The analysis in Table I clearly indicates that, in order economically to provide 150 fc, "air-water" systems utilizing water-cooled luminaries and radiant luminous ceilings are appropriate. The thermal barrier associated with the building lighting system has restricted the use of higher fc levels. Utilization of the Radiation-Convection Cooling System makes possible a shift in the position of the thermal barrier from 75 to 150 fc.

SPACE ENVIRONMENT

WINTER		SUMMER	
DB (°F)	% RH	DB (°F)	% RH
75	30	75	50

PEOPLE

INTERIOR SPACE	PERIMETER SPACE
1-1/4 PER 100 FT^2	1 PER 75 FT^2

POWER

ELECTRICAL	LIGHTING
2 WATTS/FT^2	100 FOOTCANDLES

CLIMATE REQUIREMENTS

WINTER		SUMMER	
DB (°F)	WB (°F)	DB (°F)	WB (°F)
-10	--	95	78

TYPICAL FLOOR HEATING AND AIR CONDITIONING LOAD SUMMARY

SPACE LOAD (BTU/HR-FT^2)

ITEM	N.E. PERIMETER(1)	N.W. PERIMETER	S.E. PERIMETER	S.W. PERIMETER	INTERIOR(2)
EXTERIOR HEAT LOSS	45	45	45	45	--
MAXIMUM EXTERNAL HEAT GAINS					
A. TRANSMISSION AND SOLAR	54.1	45.5	63.2	70.6	--
B. SOLAR	33.2	33.2	59.2	57.7	--
INTERNAL HEAT GAINS					
A. LIGHTING(3)	28.0	28.0	28.0	28.0	28.0
B. EQUIPMENT(4)	6.8	6.8	6.8	6.8	6.8
C. PEOPLE (5)					
1. SENSIBLE	3.0	3.0	3.0	3.0	3.0
2. LATENT	2.7	2.7	2.7	2.7	2.7
TOTAL SPACE COOLING LOAD	74.6	86.0	103.7	111.1	40.5
MAXIMUM SPACE SENSIBLE LOAD	71.9	83.3	101.0	108.4	37.8

(1) 1,300 FT^2
(2) 7,060 FT^2
(3) 100 FOOTCANDLES
(4) (2 WATTS/FT^2)
(5) (1 PERSON/75 FT^2)

Fig. 9 Occupancy requirements

DESIGN EVALUATION OF INTEGRATED SYSTEM

To provide a basis for evaluating performance of the integrated system, a comparison study of a 15-story office building was made. The 15-story office building utilizing the radiant air system and high pressure conduit systems was a 120-ft, square building, designed on the basis of a 6-ft module. The building was a free standing, steel frame structure, sheathed with an insulating glass curtain wall. An interior masonry core provided utility rooms, toilet rooms, stairways, vertical duct shaft space, piping shaft space and high speed elevators. The building consisted of a cellar, lobby, mezzanine, mechanical equipment floor and 12 typical floors, each with approximately 12,500 sq ft of rental area.

This building was of the contemporary type and utilized a 100% glass facade and a luminous type ceiling. The building was designed originally with high pressure conduit air conditioning systems, namely: a single duct induction system serving all building perimeter areas and dual duct system serving all building interior areas. In the study all floors were considered "typical" to provide a basis for system comparisons. Only areas involving movable type partitions which effectively demonstrated the design flexibility of each type of system were considered.

The economic analysis illustrating combination of the Radiation-Convection Cooling System with cellular steel floor was prepared under contract with the Inland Steel Products Co.

SYSTEM ANALYSIS

In order to compare the two systems it was necessary to evaluate all the factors which influenced the annual owning and operating costs of the systems. These items were divided into two parts, those covered under capital investment, i.e., first cost, and those associated with total energy demands which demonstrated the difference in operating cost. The cost of the equipment was obtained from actual installation data and included actual costs of all necessary equipment and installation of each system. See Fig. 9 for building occupancy re-

ITEM	FIRST COST ($ DOLLARS PER SQUARE FOOT)	
	HIGH PRESSURE CONDUIT SYSTEM	RADIATION-CONVECTION COOLING SYSTEM
MECHANICAL SYSTEM[1]	5.10	4.24
CELLULAR FLOOR AND CEILING CONSTRUCTION	8.75	10.42[2] 9.06[3]
COMBINED COST	13.85	14.66[2] 13.30[3]

NOTE: (1) EXTERNAL TO CONDITIONED SPACE
(2) TYPE I R CELLULAR FLOOR
(3) TYPE II R CELLULAR FLOOR

SEE APPENDIX A-2 FOR MORE COMPLETE ITEMIZED BREAKDOWN OF FIRST COSTS.

Fig. 10 System first cost comparison

quirements and space load analysis.

The installed refrigeration tonnage required, using the conventional high pressure conduit systems, was 1023 ton. The corresponding refrigeration tonnage, using the integrated system, was 829 ton. The relative first cost of the two building systems is given in Fig. 10.

The economic analysis for the comparison building study indicated the following savings:

Total ductwork savings 40.0%
Total mechanical savings .. 17.0%
Total building volume saved at 100 fc 5.0%
Total mechanical equipment space saved 14.7%

The use of cellular floor for supply and return air, as illustrated with the integrated system, permitted savings of approximately 5% of the building floor-to-floor height. In addition, a reduced quantity of circulated air was required as contrasted with comparable high pressure conduit systems. This item alone permitted a savings of approximately 47% in shaft space, thus increasing the availability of rentable floor area. The integrated system was found to be capable of handling up to 400 fc of illumination in less building volume than required for a comparable high pressure conduit system designed for 100 fc. A more complete breakdown of costs is given in Appendix A-2.

SYSTEM OPERATION CHARACTERISTICS AND ENERGY DEMANDS

To compare different systems in such a way as to demonstrate a month-by-month energy demand so that differences in the operation of each system were compared during different seasons of the year, a specific climate was selected using monthly average outside air, wet and dry bulb temperatures (see Fig. 11) and maximum coincident sun load to establish the probably maximum steam and electrical demand of each month. The character of this relationship from month-to-month throughout different seasons of the year, illustrated in Fig. 11, shows the significant operational features of each system in responding to the same cooling load.

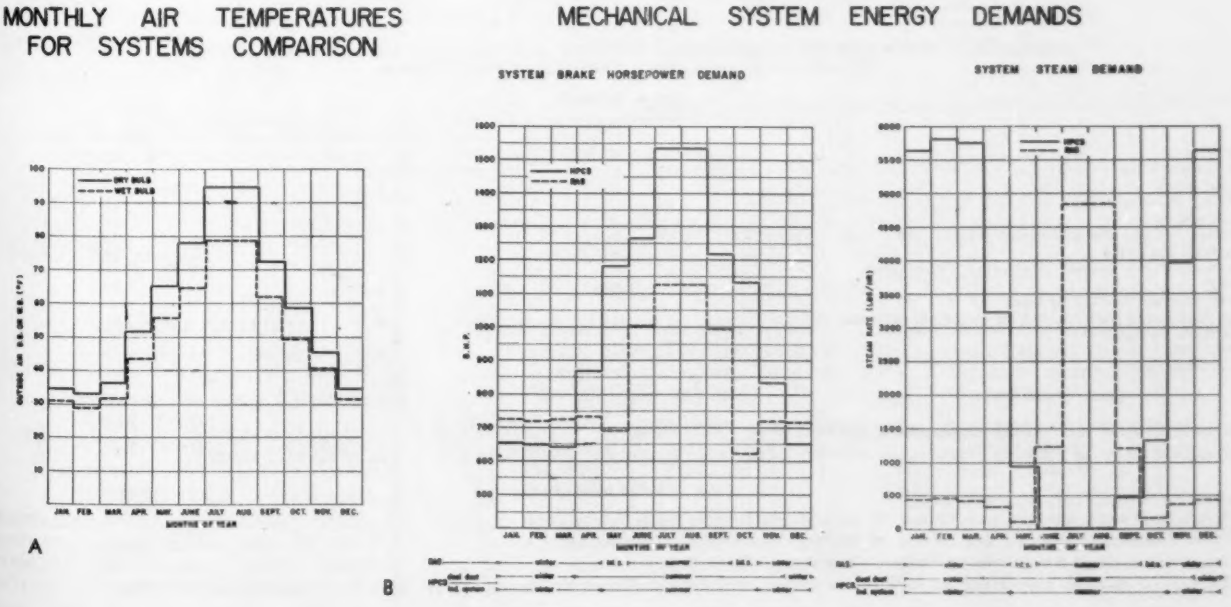
Operations Summary—1. The integrated system operated approximately 50% of the year on "Winter Cycle" and 16.7% of the year on "Intermediate Cycle," compared with induction system which operated approximately 33% of the year on "Winter Cycle."

a. For the six (6) actual months in which integrated system operated in "Winter Cycle" the average net increase in system bhp demand over high pressure conduit systems = 0.8%, while the average net savings in system steam demand over high pressure conduit system = 92.1%.

b. For the two (2) actual months in which integrated system operated in "Intermediate Cycle," the average net savings in system bhp demand over high pressure conduit systems = 43.5%, while the approximate net savings in system steam demand over high pressure conduit systems = 88%.

2. The integrated system operated approximately 33.3% of the year on "Summer Cycle," compared with the induction system which operated approximately 50% of the year on "Summer Cycle," and the dual duct system which operated approximately 67% of the year on "Summer Cycle."

Figs. 11A and 11B A—Monthly air temperatures for systems comparison. B—Mechanical system energy demands



a. For the four (4) actual months in which the radiant air system operated in "Summer Cycle," the average net savings in system bhp demand over high pressure conduit systems = 23.5%.

b. The maximum steam demand for radiant air system during any one month (July) was but 86.3% of the maximum steam for the high pressure conduit system during any one month (January).

The operating demand savings with radiant air system is summarized below:

Total savings in system bhp	26.8%
Total savings in refrigeration bhp	28.7%
Total savings in fan bhp	28.7%
Total savings in pump bhp	3.9%
Reduction in installed motor hp	21.8%

Refer to Appendix A-3 for a more complete breakdown.

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2. "Heat Redistribution Systems", by G. Meckler (Progressive Architecture—December, 1959).
3. "Is Humidity Important in the Temperature Comfort Range", by Koch, Jennings and Humphreys (ASHRAE JOURNAL — April, 1960).
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APPENDIX A-2

MECHANICAL—FIRST COST

	High Pressure Conduit System			Radiant Air System		
Refrigeration Equipment	1023		$105,600	829 ton		$ 85,580
Ductwork						
Typical Floor (due to use of Cellular floor)	11730 lb x $.75 x 12 2460 lb x $.85 x 12	$105,570 25,092		5446 lb x $.75 x 12		49,014
			130,662			
Shaft	32506 lb x $.90	29,255		34695 lb x $.80	$ 27,756	
14th Floor	44905 lb x $.90	40,414		46439 lb x $.85	39,473	
Lobby		7,500			7,500	
Mixing Boxes		15,000				
Grilles	125 x 12 x $7.00	10,500		125 x 12 x $7.00	10,500	
Diffusers	164 x 12 x $20.00	39,360				
Flexible Connection	698 ft x $1.25 x 12	10,470		738 ft x 12 x $1.25	11,070	
Continuous Grille				960 ft x 12 x $2.00	23,040	
			$152,499			$119,339
Air Handling Equipment			82,500			60,500
Perimeter Panel						30,000
Dehumidification						60,000
Induction Units			48,000			
Sill Enclosures			39,900			
Steam Piping			35,805			29,015
Controls			102,300			99,480
Chilled Water						
Floors		$118,162			$151,792	
Basement		45,000				
14th Floor		113,545			108,878	
			$276,707			$260,670
Cooling Tower			31,000			31,000
Condenser Water			23,690			30,590
	Grand Total		$1,028,663			$855,188
Mechanical—First Cost/SF			$ 5.10/SF			$4.24/SF

SF figures based on a total of 201,600 SF (12 Typical Floors, Basement and 14th Floor Mechanical Room).

ARCHITECTURAL SAVINGS

Structural Steel — 27 Columns at 0.67 ft x 12 =	217 SF × 150 lb = 32,550 lb at $400.00/ton	$ 6,508.00
Core Material (Concrete Blocks + Finish)	540 ft × 0.67 ft × 12 = 4342 SF at $1.00/SF	4,342.00
Exterior Wall (Curtain Wall)	480 ft × 0.67 ft = 322 × 12 = 3864 SF at $10.00/SF	38,640.00
Shaft Space (47% savings with radiant air system)	224 SF × 12 = 2688 SF at $25.00/SF	67,200.00
Machine Equipment Space (14.7 savings with radiant air system)	65 ft × 31 = 2015 SF at $10.00/SF	20,150.00
	Grand Total	$136,840.00

COMPARISON BETWEEN HIGH PRESSURE CONDUIT SYSTEM AND RADIANT AIR SYSTEM CELLULAR FLOOR AND CEILING CONSTRUCTION

	High Pressure Conduit System	Radiant Air System	
		I-R Cellular Floor	II-R Cellular Floor
Floor Tile	$ .95/SF	$ .95/SF	$ .95/SF
Concrete Topping	.21	.21	.19
Cellular Deck	.22	2.32	1.20
Electrical Raceway System	.40	.40	.40
Fire Proofing	.30	.35	.13
Ductwork (due to use of cellular floor)	.89	.33	.33
Acoustical Material	.28	.14	.14
Fixture-Raceway System (including upper tees)	2.40	—	—
Wiring	1.00	1.10	1.10
Louver Ceiling (including tees)	1.10	—	—
Upper Radiant Panel (including electrical components)	—	2.65	2.65
Suspension System	—	.12	.12
Lower Grid	—	1.35	1.35
Lower Grid Tees	—	.50	.50
	$ 8.75/SF	$10.42/SF	$ 9.06/SF

Total cellular floor and ceiling construction addition using I-R Cellular Fl. 10.42-8.75 = 1.67 × 146592 SF = $244,808.00
Total cellular floor and ceiling construction addition using II-R Cellular Fl. 9.06-8.75 = 0.31 × 146592 SF = $ 45,443.00

SUMMARY

(1) Total duct work savings due to use of radiant air system (40% Savings)	$114,808.00
(2) Total mechanical savings due to use of radiant air system (17% Savings)	173,475.00
(3) Ductwork savings due to use of radiant air system cellular floor	81,648.00
(4) Mechanical savings due to use of radiant air system excluding ductwork savings due to use of radiant air system cellular floor	91,827.00

(5) Total Architectural savings $136,840.00.
(6) Total cellular floor and ceiling construction addition using I-R cellular floor (including ductwork due to use of cellular floor) $244,808.00
(7) Total cellular floor and ceiling construction addition using II-R cellular floor (including ductwork due to use of cellular floor) $45,443.00
Total addition using I-R cellular floor: (items 6 — (4 + 5) = $16,141.00
Total savings using II-R cellular floor: (items 4 + 5 — 7) = $183,224.00

APPENDIX A-1

The derivations of Equations (1) and (2) presented in Fig. 6 of this paper express the design relationships necessary to relate the design outlet air conditions leaving the chemical air conditioner with the humidity conditions within the space. The derivation of the general relationships are given below:

Total occupancy humidity control is accomplished by humidifying or dehumidifying outside air. Thus, we can write:

$$(G_s - G_{os}) = (G_w - G_{ow}) = L_d/0.68\ C_{os}$$

Derivation of Equation (1)—Referring to the psychrometric at 75 F/50% RH: $G_s = 65$

Therefore:

$$(65 - G_{os}) = L_d/0.68\ C_{os}$$

Rearranging:

$$G_{os} = 65 - L_d/0.68\ C_{os} = [44.2\ C_{os} - L_d]/0.68\ C_{os}$$

Similarly for the winter design space condition at 75 F/30% RH: $G_w = 39$

Therefore:

$$(39 - G_{ow}) = L_d/0.68\ C_{os}$$

Rearranging:

$$G_{ow} = 39 - L_d/0.68\ C_{os} = [26.5\ C_{os} - L_d]/0.68\ C_{os}$$

Derivation of Equation (2)—Let G_m = absolute humidity of total air mixture entering conditional space (grains moisture/lb/dry air)

When the occupancy zone is at a peak latent load condition all of the zones are assumed to be operating at the design point (75 F/50% RH). Therefore, we can write the following equation:

$$G_{os}\ C_{os}/C + (C - C_{os})\ G_s/C = G_m \qquad (a)$$

But if G_{ms} is not to be exceeded:

$$G_m = G_{ms} - L_d/0.68\ C = [0.68\ C\ G_{ms} - L_d]/0.68\ C \qquad (b)$$

Therefore, we can equate (a) and (b) above as follows:

$$G_{os}\ C_{os}/C + (C - C_{os})\ G_s/C = [0.68\ C\ G_{ms} - L_d]/0.68\ C$$

For the condition of $L_d = L_m$

$$G_{os}\ C_{os}/C + (C - C_{os})\ G_s/C = [0.68\ C\ G_{ms} - L_m]/0.68\ C$$

Multiplying through by (C):

$$G_{os}\ C_{os} + G_s\ C - G_s\ C_{os} = G_s\ C - 1.47\ L_m$$

Rearranging and multiplying by (—1):

$$1.47\ L_{ms} + G_s\ C - (G_s - G_{os})\ C_{os} = G_{ms}\ C$$

For summer space design from psychrometric chart:

$G_s = 65$ $\qquad$ $G_{ms} = 78$

Therefore, we can write:

$$1.47\ L_{ms} + 65\ C - (65 - G_{os})\ C_{os} = 78\ C$$

Rearranging $L_{ms} = [(65 - G_{os})\ C_{os} + (78 - 65)\ C]/1.47$

Similarly for winter space design (75 F/35% RH)

$G_w = 39$ $\qquad$ $G_{mw} = 46$

Therefore, by similar development, we can write:

$$L_{mw} = [(39 - G_{ow})\ C_{os} - (46 - 39)\ C]/1.47$$

APPENDIX A-3*

COMPARISON OF SYSTEM BHP ASSOCIATED WITH 12 TYPICAL FLOORS

(Summer Design Condition)

	High Pressure Conduit System	Radiant Air System
Refrigeration Machines	962	687
Fans	397	254
Pumps	177	184
Total operating bhp	1536	1125

COMPARISON OF BUILDING ELECTRIC MOTOR DEMAND

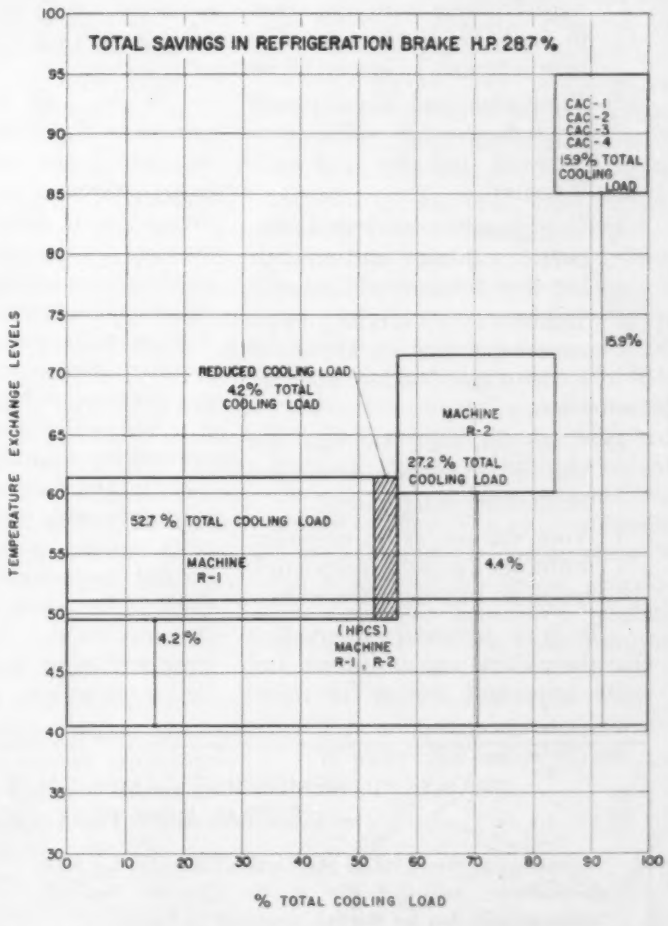
	High Pressure Conduit System	Radiant Air System
Total Bldg. (All Floors) installed motor hp:	1754.5	1371.5

RADIANT AIR SYSTEM — SUMMARY OF RESULTS

A. Net savings in overall system bhp = 26.8%
B. Net demand high pressure conduit system bhp/cooling ton* = 1.60 (1.49 kw/ton)
C. Net demand radiant air system bhp/cooling ton* = 1.16 (1.08 kw/ton)
D. Net savings in system refrigeration bhp = 28.7%
E. Net savings in system fan bhp = 36.0%
F. Net increase in system pump bhp = 3.0%
G. Net savings in Bldg. installed motor hp utilizing radiant air system = 21.8%

* Based on 962 ton high pressure conduit system load for all typical floors.

* A design comparison of a 15-story building was made to compare the use of Type IR and IIR cellular air floor arranged with the integrated radiation-convection system with conventional high pressure conduit systems using ductwork distribution.



NOMENCLATURE

L_d = Occupancy design latent load (Btu/hr/ft²) which is taken the same for summer and winter

C_{os} = cfm/ft² of outside air circulated

C = cfm/ft² total air circulated

G_{mw} = Absolute humidity (maximum allowable) in winter conditioned space (grains moisture/lb dry air)

G_{ms} = Absolute humidity (maximum allowable) in summer conditioned space (grains moisture/lb dry air).

G_{os} = absolute humidity leaving chemical air conditioner during summer operation (grains moisture/lb dry air)

G_{ow} = absolute humidity leaving chemical air conditioner during winter operation (grains moisture/lb dry air).

G_{mw} = absolute humidity (maximum allowable) in winter conditioned space (grains moisture/lb dry air)

L_{mw} = maximum allowable occupancy design winter latent load (Btu/hr/ft²)

G_s = absolute humidity at summer design space relative humid-

(*Continued on page* 106)

Is ASHRAE the Place for Me?

Several years ago our Society compiled a long list of compelling reasons why one should become a member, among which were the following:

A Personal contacts at National and Chapter levels of the new and established engineers

B Society source of technical information in discussions, talks, publications, factual data

C Stabilizing industry influence, favorably accepted in peace and troublesome times

D Acquaintance with engineers in many fields, through common interests in every clime

E Promotion and development through research within government, industry and individuals

F Inauguration of standards, codes, seminars and unfolding new horizons collectively

G Industry's outstanding engineers are the backbone of society membership and action

H Contributions to the art through numerous diversified committee assignments

I Your stature, as an engineer, enhanced and supported, publicly and industry wide.

It is a personal observation that there exists one additional and quite important reason for membership in ASHRAE–IT IS THAT YOUR MEMBERSHIP IS PERSONAL.

This society recognizes no group, corporation or company memberships, nor the subsidizing of dues for any individual applicant or member. Your business affiliation has no bearing on your acceptance or assignment to any committee. You carry your membership, in your name, to the address you select, wherever you go; just as long as you hold that privilege high it will not be revoked. In addition your membership is universal, honored in every country where you will find also many mutual contacts. Never permit it to lapse.

When, and if, you are approved by the Admissions Committee, where the writer served for many years, and you have expressed your desire to become a member by thoughtful answers regarding your background, interests, education, references and the fact that you might prove to be a worthy member, you, as an individual, will get every consideration.

You will have numerous opportunities to make this a better society. You may not be impressed favorably with certain internal society actions, you may feel they do not meet your standards, this, then, is the place and the time to become vocal. Attend the meetings and after careful consideration, present your convictions until you sway those occupying the chairs, or gathering the opposition ideas you may convince yourself that the program actually has some merit.

C. R. LOGAN
Presidential Member ASHRAE

As a society we've made mistakes, even as you or I, but ASHRAE has, and can fill even a greater need, and you may be the very member who can bring this about.

George Washington's admonition may prove convincing, if you are still in doubt –
"ASSOCIATE YOURSELF WITH MEN OF GOOD QUALITY IF YOU ESTEEM YOUR OWN REPUTATION."

MEMBERSHIP CAMPAIGN IS OFF TO A FLYING START — CENTRAL NEW YORK CHAPTER LEADS THE RACE —

Acceptance of the ASHRAE Membership Campaign has been demonstrated enthusiastically at the Chapters Regional Meetings held thus far this fall, especially in Region I.

This Region has produced the first winner . . . CENTRAL NEW YORK, which has increased its net membership by 14.3%, exceeding its quota and winning the Membership Development Committee's first citation.

In second place, from Region VII, is Middle Tennessee.

Many chapters have not yet submitted reports on their first quarter's progress. We hope to hear from all chapters as soon as possible so that a comprehensive analysis can be presented in the March issue of the ASHRAE JOURNAL.

The results to date have been gratifying. However, we urge all chapters to make every effort to achieve their quotas as quickly as possible. Keep up the good work and give Central New York and Middle Tennessee a run for their money!

W. H. MULLIN, Chairman
Membership Development Committee

Every Member an ASHRAE Booster

Design of a water type

Carbon Dioxide Absorption

system for fruit storage

The controlled atmosphere (CA) method of fruit storage as utilized in the United States for apples and pears requires a means for removing the excess quantities of carbon dioxide produced by the fruit. Since a low oxygen level, usually 3%, is needed, the carbon dioxide removal must be accomplished without the addition of large quantities of oxygen. Caustic soda (NaOH) commonly is used for chemical absorption of carbon dioxide.[13, 3, 9, 16] In Europe, milk of lime ($Ca(OH)_2$) systems are in common use.* The mono-, di- and tri-ethanolamines have been used under experimental conditions[11] for removing carbon dioxide from CA storages; a mono- or tri-ethanolamine scrubber has been developed and is used in England.[7] Palmer[8] discovered that it was possible to remove carbon dioxide from CA storages using a water absorption-desorption system.

Several types of water carbon dioxide scrubbing systems are in use in New York State[14, 15, 4] in which the carbon dioxide is absorbed in the water circulated over a wet type refrigeration evaporator and removed from this water in an aeration tower located outside the CA storage. A similar absorption system using a spray desorption system has been developed in California.[17]

This report describes the design and operating characteristics of a single tower water absorption-desorption system for removing carbon dioxide from CA storages. Since this system operates independently of the refrigeration system it is classified as a separate

I. J. PFLUG

absorption system, as differentiated from systems that utilize the refrigeration evaporator as an integral part of the absorption system.

A water absorption system which operates independently of the refrigeration system is needed where dry type refrigeration evaporators are used. It is also desirable in a CA room operated at 31-32 F since a wet coil would require an antifreeze material in the water, which significantly reduces the carbon dioxide absorption efficiency of the system.

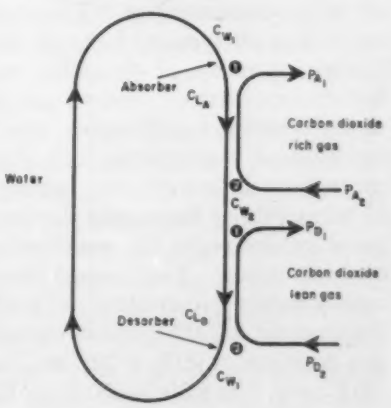
OPERATING PRINCIPLES

The principles of operation of the water type carbon dioxide absorption-desorption system are entirely different from those of the caustic soda (NaOH) type carbon dioxide absorber that it replaces. The caustic soda system absorbs and removes carbon dioxide from the CA storage room by chemically reacting with the carbon dioxide. It removes carbon dioxide at an exceedingly high rate when the unit is freshly charged and at a notably low one when the charge is nearly exhausted and the solution is ready for discarding. In an emergency, with recharging every two or three hr, a caustic soda absorber can be used to remove 3 to 5 times more carbon dioxide.

In the water absorption system carbon dioxide moves out of the CA storage room by a 2-part diffusion process in which water is the absorbing solution in both parts of the system. The transfer of carbon dioxide by water from the CA storage can be explained with the aid of the schematic diagram in Fig. 1. Starting at the top of the tower, water flows downward through the absorption section where it comes into contact with air from the storage, which may contain, for example, 5.0% carbon dioxide. Some of the carbon dioxide diffuses into and is absorbed by the water. The water next flows through the desorbing section where it comes into contact with normal atmosphere that may have but 0.03% carbon dioxide and where some of the carbon dioxide is desorbed from the water and diffuses out of the water into this low carbon dioxide atmosphere. The water normally is recirculated for continuation of the absorption-desorption process.

The mass transfer of carbon dioxide to or from a water solution

Fig. 1 Schematic diagram of water-type carbon dioxide absorption system



* Michigan Agricultural Experimental Station Journal Article No. 2780.

I. J. Pflug is Professor of Food Science, Michigan State University. The paper was prepared for presentation at the ASHRAE Semiannual Meeting, St. Louis, Mo., January 29-February 1, 1962.

is liquid-film-controlled,[6,12] which can be described by the equation:

$$N = K_{La} V \Delta C$$

where: N is the lb-moles of solute transferred per hr

K_{La} is the overall capacity coefficient,

$$\frac{\frac{\text{lb-moles}}{\text{cu ft hr}}}{\frac{\text{lb-moles of solute}}{\text{cu ft/soln}}}$$

V is the tower volume, cu ft

ΔC is the log mean concentration difference

$$\frac{\text{lb-moles of solute}}{\text{cu ft/soln}}$$

The diffusion coefficient, liquid viscosity, temperature and liquid flow rate determine the mass transfer coefficient K_{La} for carbon dioxide in water. Liquid flow rate is the most important of these variables; the mass transfer coefficient varies approximately directly with liquid flow rate.[5] Both the diffusion coefficient and viscosity are temperature dependent; increasing the temperature increases the diffusion coefficient and reduces viscosity. Therefore, the mass transfer coefficient is increased by an increase in the diffusion coefficient and a decrease in the liquid viscosity. Under usual conditions, gas flow rate does not significantly affect the absorption coefficient.

The carbon dioxide concentration difference at any point in a liquid-film-controlled system is the difference between the carbon dioxide concentration in the bulk of the liquid and the equilibrium liquid concentration of the carbon dioxide in the gas phase expressed as molar concentration. The concentration difference of the system is the log mean of the inlet and outlet conditions. Since carbon dioxide-water equilibriums are a function of temperature the concentration difference is a function of temperature; increasing the temperature decreases the equilibrium concentration. The equilibrium concentration[1] (expressed as mole fraction) of a 5.0% carbon dioxide gas in water is 3.08×10^{-5} at 77 F (25 C) and 4.88×10^{-5} at 50 F (10 C). Therefore, the change in absorption rate due to a change in temperature is quite small since the increase in K_{La} brought about by an increase in temperature is offset by the decrease in concentration difference.

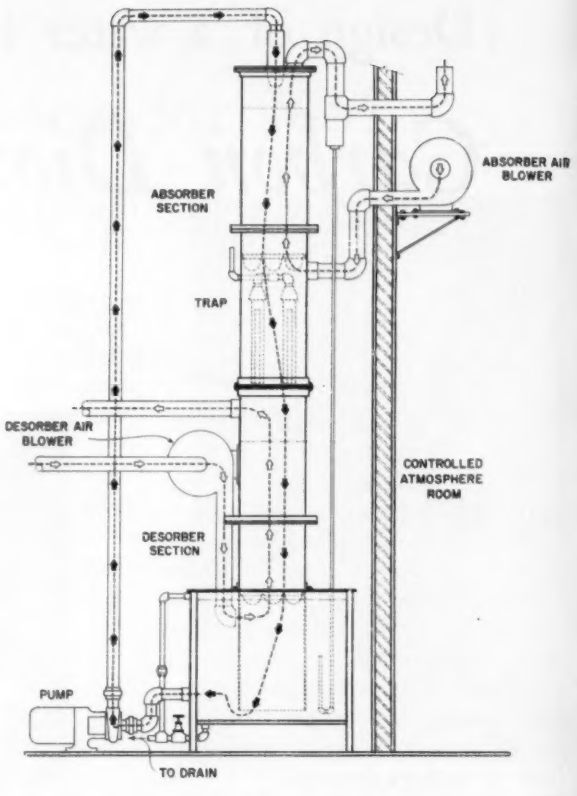


Fig. 2 Cross section diagram of water absorber for CA fruit storage

In general, the carbon dioxide level of the CA storage is the largest factor in determining the carbon dioxide concentration difference, and therefore the carbon dioxide removal rate for a specific system.

The quantity of carbon dioxide removed from a CA storage by a water absorption-desorption system increases linearly with packing depth; however, 4.5 ft is the approximate maximum economical packing depth for both the absorption and desorption sections.

CONSTRUCTION OF SINGLE TOWER ABSORBER-DESORBER UNITS

The single tower type absorber illustrated in Fig. 2 consists of absorber section, trap section, desorber section, reservoir tank, liquid pump, absorber air-blower, desorber air-blower and piping. One-in. porcelain saddle packings* are used in both the absorber and desorber sections. There is a single liquid circuit; the liquid from the reservoir tank is pumped into the top of the unit; it then flows down through the absorber section packing, through the trap, through the desorber section packing, and finally into the reservoir tank.

There are two gas circuits: the CA storage room gas circuit, which moves gas from the absorber gas-blower located in the room to the bottom of the absorption section, up through the packing and then back into the room; the low carbon dioxide gas circuit, which consists of flow of outdoor air from the blower into the bottom of the desorber section, up through the desorber packing and discharges outdoors.

Correct construction of the liquid trap located between the absorber and desorber sections is essential for successful operation. Water flow from the absorber section to the desorber section must take place without removal of CA storage room gas from the absorber section or the addition of outside air from the desorber section into the absorber section. A diagram of a section of the trap is shown in Fig. 3. This trap design has a capacity of approximately 18 gpm per pipe unit. A complete trap will consist of 4 to 9 units, depending upon the size of the system.

Several sizes of absorption sys-

* Manufactured by the U. S. Stoneware Co.

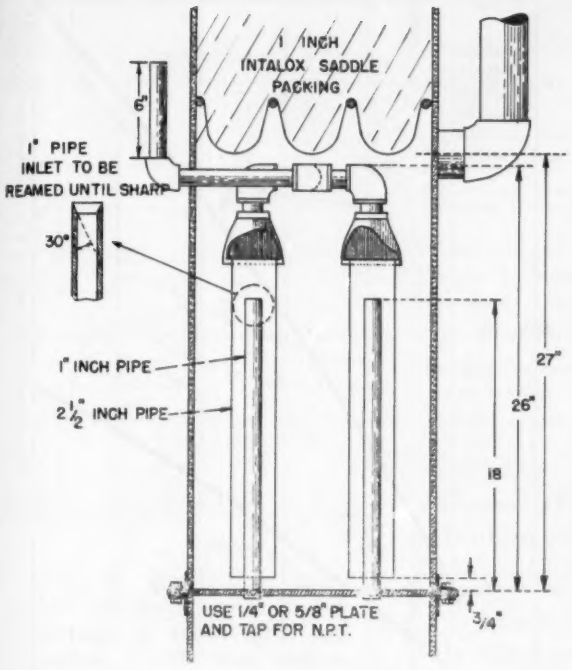


Fig. 3 Trap that allows water to flow from absorption section to desorption section without gain or loss of storage room gas

tems are in use at the present time: 15 x 15-in. square towers with a 5-pipe trap; 16 x 20-in. rectangular towers with a 6-pipe trap; 16 x 30-in. towers with an 8-pipe trap; and 24 x 24-in. towers with a 9-pipe trap.

A packing depth of 4.5 ft in both the absorption and desorption sections is the approximate economic maximum for CA storage carbon dioxide removal conditions. This depth therefore will give the greatest carbon dioxide removal rate for a given carbon dioxide level and liquid flow rate. Since reservoir, trap and distributor requirements are 3 ft, 2.5 ft and 2 ft, respectively, (one ft for absorber and one ft for desorber) the minimum over-all height of a unit with 4.5 ft of packing is 16.5 ft.

In commercial units designed to date, a 3-ft packing depth has been used in both absorber and desorber, which gives an over-all unit height of about 13.5 ft.

A fractional hp centrifugal pressure blower with capacity of 50 cfm of gas per sq ft of tower at a pressure of 4.0 in. of water gage is required for both the absorber and desorber sections.

TESTING A PILOT-SIZE WATER ABSORPTION SYSTEM

A pilot-size water absorber-desorber system was constructed out of 8-in. OD (7¾ ID) plastics pipe. Both the absorber and desorber sections contained 36 in. of 1-in. saddle packing. This unit was set up in the laboratory where it was possible to measure and control the air flow through the absorber section, air flow through the desorber section, and liquid flow. A closed system having a total volume of about 4000 liter was used to maintain and supply a carbon dioxide rich atmosphere which was pumped from the room through the absorption section of the tower and back into the room. As the gas returned to the room it passed through a tee where carbon dioxide was added at a continuous rate.

The system was operated until steady state conditions were established prior to making a test. The room containing the carbon dioxide rich atmosphere was refrigerated; the air supplied to the desorber was passed through a refrigerated chamber so that it was at approximately the recirculated water temperature when it entered the desorber. The temperature of the recirculating water was controlled.

The carbon dioxide level in the room was determined using an Orsat type volumetric gas analyzer. The smallest division on the instrument used was 0.10%; the gas level could be estimated to the nearest 0.05%.

The procedure followed in making a test was to allow the system to equilibrate, during which time regular gas analysis was made until the carbon dioxide level in the system remained constant. The tests then were started and required approximately 30 min to complete. At the beginning of each 5-min period a sample of the circulating water was taken as it left the absorbing section and as it left the desorbing section. At the same time, the carbon dioxide level, the gas flow rate and the liquid flow rate were measured. Carbon dioxide was added to the room using a rotometer as a control device plus a wet test gas meter as a more precise measure of the amount of carbon dioxide added. Temperatures at 6 points in the system were recorded during the test period. Temperatures were measured using copper-constantan thermocouples and an appropriate temperature measuring potentiometer.

The samples collected as the water left the absorption section and as the water left the desorbing section were analyzed chemically for carbon dioxide, as follows: Approximately 0.01 normal sodium hydroxide and 0.01 normal hydrochloric acid were prepared and standardized. Prior to starting the test, 75 ml of the standard sodium hydroxide solution was added to each of twelve–250-ml volumetric flasks. The sample of absorption fluid was deposited directly into the sodium hydroxide solution in the 250-ml flasks until the flasks contained 250 ml. The contents of the flasks were mixed by shaking a 50-ml aliquot was removed and potentiometrically titrated with standard hydrochloric acid to the carbonate-bicarbonate end point.

Distilled water was used as the absorbing medium in the test. The packing was boiled in both alkali and acid cleaning solutions prior to use and then washed thoroughly using distilled water before actual testing was started. It was assumed that the normal content of the air was 0.03% carbon dioxide.

The data from the 6 replications of each test were evaluated individually; the resulting K_{La} values were averaged to give a K_{La} for the test. It was assumed that the K_{La} for the absorption section and the K_{La} for the desorption section were equal and since the amount of carbon dioxide absorbed was equal to the amount of carbon

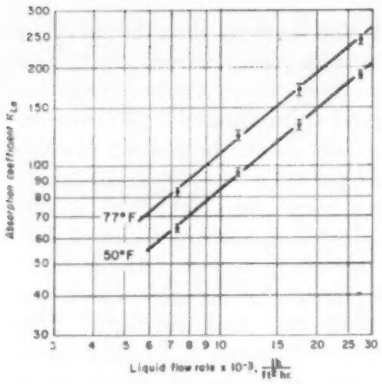


Fig. 4 Absorption coefficient K_{La} for 1-in. saddle packing as a function of water flow rate

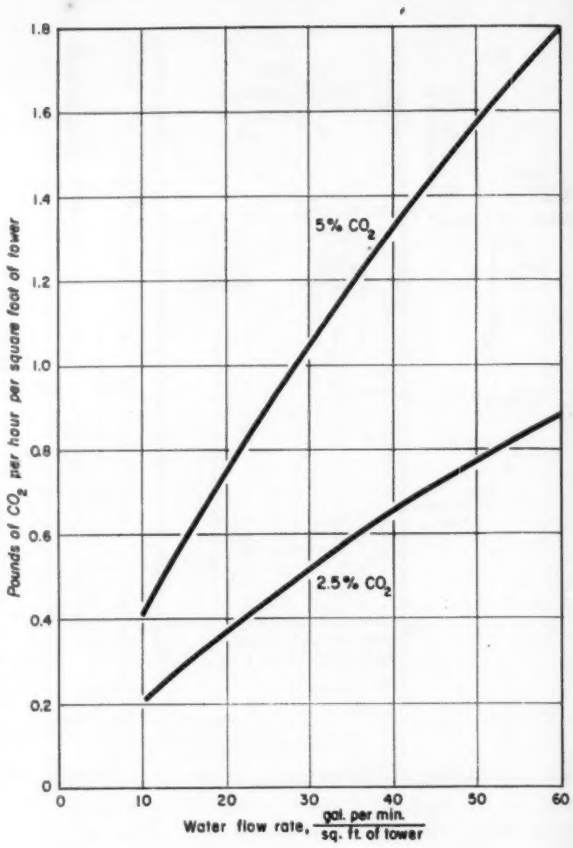


Fig. 5 Carbon dioxide removal capacity using 3 ft of packing in both absorber and desorber

dioxide desorbed and the volume of packing was the same in both the absorber and desorber sections, the concentration difference in the absorber section had to be equal to the concentration difference in the desorber section. This premise made it possible to eliminate the problem of differentiating between carbonate ion, bicarbonate ion and absorbed carbon dioxide in the calculation of the concentration difference.

The effect of flow rate on absorption coefficient K_{La} was determined at 50 F (10 C) and 77 F (25 C) at a gas flow rate of 45.8 cfm per sq ft of tower. The effects of 30.5, 45.8 and 61.0 cfm of gas flow per sq ft of tower were compared at 77 F (25 C) for a liquid flow rate of 27,500 lb/ft² hr.

The results of the tests to determine the effect of liquid flow rate on K_{La} for a gas flow rate of 45.8 cfm per sq ft of tower at 50 F (10 C) and 77 F (25 C) are shown in Fig. 4. These results compare favorably with the results of Koch, et al. An analysis of variance of the data showed no significant difference in K_{La} when the gas flow rate was increased to 61.0 cfm per sq ft of tower; however, when the gas flow rate was decreased to 30.5 cfm per sq ft of tower the resulting average K_{La} of 219 was significantly different from the K_{La} for 45.8 and 61.0 cfm per sq ft of tower.

In Fig. 5 are shown curves relating the carbon dioxide removal capacity in lb/hr at 50 F for a gas flow rate of 50 cfm to liquid flow rate in gpm/ft² of tower area for 2.50 and 5.00% carbon dioxide levels. The data in Fig. 5 are calculated from the data in Fig. 4. The recommended design flow rate for water type carbon dioxide absorption-desorption systems is 45 to 50 gpm/ft² (22,500 to 25,000 lb/hr). The absorption capacities in Fig. 5 are for pure water.

EFFECT OF A WATER TYPE ABSORBER ON THE CA STORAGE

Since the water carbon dioxide absorption system replaces the caustic soda carbon dioxide absorption system it is important to consider and evaluate the differences in the effect of these two systems on the CA storage room.

The physical principle on which the water type carbon dioxide absorption-desorption system is based does not exclude other gases from diffusing through the system; therefore, under low oxygen CA storage conditions oxygen moves from outside air to the CA atmosphere. The absorption coefficients for oxygen in water and carbon dioxide in water are approximately equal; however, the relative solubilities of carbon dioxide and oxygen are in the ratio of 10 to 1; therefore, under equal driving force conditions, 10 times as much carbon dioxide will move out of the room as oxygen into the room. Consequently, the quantity of oxygen that diffuses into the room is not large enough to offset the advantages of a water type carbon dioxide absorption system.

An analysis of the effect of water absorption of carbon dioxide on the operation of the CA storage room[10] showed that tight CA rooms filled with fruit should require but two or three days longer to develop a 3% oxygen atmosphere than similar rooms operated with caustic soda absorbers. This analysis also showed that rooms which are borderline in tightness or partially filled will not operate satisfactorily when a water absorption system is used for carbon dioxide removal. The results of this analysis have been substantiated in the field where a large number of commercial rooms have been operated during the entire year, including the oxygen reduction period, with only a water absorption system for the removal of carbon dioxide.

SIZING THE WATER ABSORBER

Theoretically, the carbon dioxide absorption system for the CA storage room should be designed on the basis of quantity of fruit in the CA room, the carbon dioxide production rate of this fruit and the carbon dioxide level of the storage. The recommended oxygen, carbon

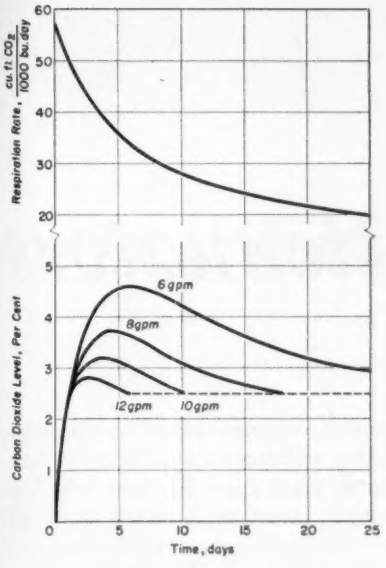


Fig. 6 Respiration curve for McIntosh apples in CA storage at about 36 F and carbon dioxide level pattern as a function of liquid flow in gpm per thousand bushels (flow rate 45 gpm/ft²)

dioxide and temperature for the five varieties of apples commonly stored in Michigan are listed in Table I.

Because the CA storage operator in Michigan does not know from year to year what variety of fruit he will store, each room must be equipped with an absorber of adequate capacity to handle the most difficult carbon dioxide removal job. This condition in Michigan and in the Northeast United States is produced by the McIntosh variety of apples which requires a temperature of 36 to 38 F and a carbon dioxide level of 2.5% during the first month of CA storage.

A carbon dioxide production rate curve for McIntosh apples as a function of days after sealing the CA storage[10] is shown in Fig. 6. The carbon dioxide level in the storage, assuming this carbon dioxide production rate, a space utilization factor of 3 cu ft per bushel and water absorption of carbon dioxide with a water flow rate of 45 gpm/ft² started 24 hr after sealing room was calculated, using a finite differences procedure[10] for absorber capacities of 6, 8, 10 and 12 gpm per 1000 bushel of fruit. The results of these calculations are plotted on the lower part of Fig. 6. A flow rate of 10 gpm per 1000 bushel is being recommended for

Table I. Recommended CA Storage Conditions for Michigan Apples

Variety	Oxygen Level (%)	Carbon Dioxide Level (%)	Temperature (F)
McIntosh	3	5[a]	36-38
Jonathan, Red Rome	3	5	32
Delicious, Red and Yellow	3	2.5	32
Northern Spy[b]	3	8	38

[a] A carbon dioxide level of 2.5% is recommended for the first month.

[b] The conditions utilized for Delicious have been used satisfactorily for Northern Spy.

Table II. Maximum bushels of apples for three sizes of water type carbon dioxide absorption systems.

Rectangular Tower Cross-Section, In.	Water Flow Rate, gpm	Bushels of apples
15 × 15	70	7,000
16 × 20	100	10,000
24 × 24	180	18,000

Michigan use in absorber-desorber systems of the type illustrated in Fig. 3.

Ideally, a water-carbon dioxide absorption system should be designed specifically for each CA storage; however, since practically no two CA storages are the same size and sizes range from 5000 to 20,000 bushels it is not economical to do this. It is more economical to fabricate three or four sizes of water absorption systems in quantity. Approximate CA storage room capacities that can be handled by 15 x 15, 16 x 20 and 24 x 24-in. water absorption systems with a liquid flow rate of 45 gpm/ft² of tower are shown in Table II.

The temperature of the CA storage is a critical factor in maintaining carbon dioxide control during the early days of the storage season; McIntosh apples will produce approximately 35% more carbon dioxide at 41 F than at 36 F. It is not economical to design a 35% safety factor into the absorption system; consequently, the fruit must be cooled to the storage temperature and maintained at this temperature.

Water-carbon dioxide absorption systems in Michigan are designed and constructed so that caustic soda can be used in an emergency.

CONTROLLING THE CARBON DIOXIDE LEVEL

It is obvious from Fig. 6 that a few days after the peak carbon dioxide level is reached there will be excess absorption capacity. At this point the carbon dioxide level is controlled by reducing the operating time of the absorption system. It has been found practical to use a timer adjustable from 0 to 20 hr maximum operating time with a hold position. The timer control circuit is connected to magnetic starters or relays operating the pump and blowers. The daily routine for carbon dioxide control consists of analyzing the CA storage room atmosphere for carbon dioxide and oxygen after which the timer is set for the hours of operation needed to maintain the carbon dioxide at the desired level. This system has been found to be an ideal solution to the carbon dioxide control problem.

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Noise reduction characteristics of

Package Attenuators

My original paper on this subject (see footnote) contained data concerning commercial package attenuators manufactured by three companies. Following the publication of this paper, the Sales Manager of one of these organizations wrote a lengthy letter protesting numerous points concerning the data portion of the paper. Representatives of that company presented data taken in their laboratories using a measurement technique which differed from mine.

These data, (Fig. 1, curve 1) showed a much higher noise reduction than the values reported in my paper (curve 2). No errors in my original data were found which could account for the large discrepancies. As I noted subsequently in a Letter to the Editor published in the May 1960 ASHRAE JOURNAL, " . . . this discrepancy is sufficiently large that we feel it represents a major technical dispute which can be settled only by intercomparing the two test techniques."

We anticipated the tests would involve comparison tests of the identical muffler first in their laboratory, using their technique, and second in our laboratory, using our technique. This point seemed important as we had not tested the identical muffler, nor had we even tested mufflers of the same size.

The testing program we undertook became more elaborate than anticipated because measurement of an identical unit in both of our laboratories was not possible. One test setup is designed for 2 x 2-ft mufflers and was ill-suited to measure smaller mufflers. On the other hand, our test setup is designed for mufflers having a maximum cross section of 1 x 2 ft, although we have attempted measurements on units up to 14 in. x 2 ft.

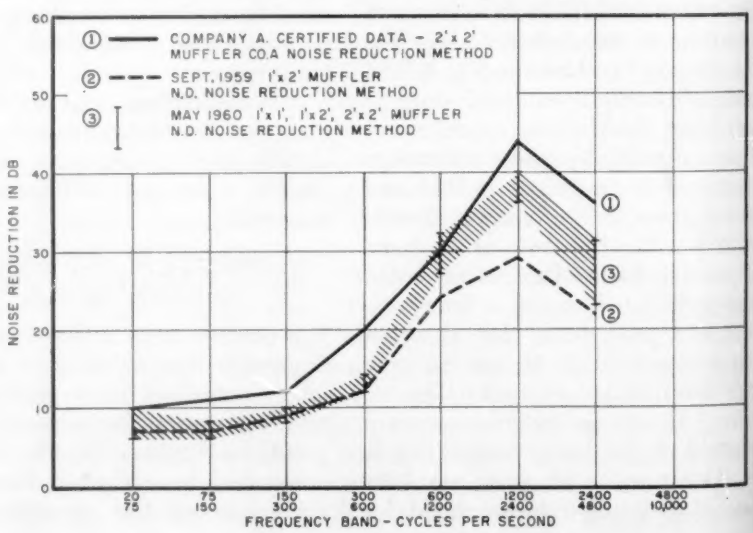
NORMAN DOELLING

We undertook therefore to compare the numbers obtained using the two noise reduction techniques. The techniques are similar. They differ in that the one described in my paper uses five microphone positions in a plane in front of, and in a plane behind, the muffler. Theirs uses a single position in each case. Further, my "up stream" position is about 6 in. in front of the muffler, while theirs is about 6 ft from the muffler.

We tested a 2 x 2-ft Unit A-5 in the laboratories of Company A using the two test methods. These tests also were observed and recorded with a different "read-out" system by Lewis S. Goodfriend of Lewis S. Goodfriend and Associates, who was retained by Company A. The results obtained in comparing the two methods are given in lines A.1 through A.3 of Table I. Since the identical muffler was used (and was not moved) in the identical test duct, the differences result only from differences in test technique. The measurement technique which I used yields the lower numbers. Mr. Goodfriend's reduction of the data showed an even larger difference between the two techniques.

I believe my method (which was used for **all** of the data in my

Fig. 1 Noise reductions of several Model B mufflers using two noise reduction techniques (muffler A-5 in original paper)



Norman Doelling, who authored "Noise reduction characteristics of packaged attenuators for air-conditioning systems," as presented at the ASHRAE Dallas Semiannual Meeting, February 1-4, 1960, is Senior Consultant with Bolt, Beranek and Newman, Inc. The original paper appeared in the ASHRAE JOURNAL of February 1960.

for air conditioning systems

— an addendum

paper) is to be preferred as more realistic and significant for users of mufflers. Some additional work which has been carried out in part by me and other members of an American Standards Association committee concerned with the problem of measuring mufflers suggests an outline of a third technique which may well be even more realistic and correlate even better with field experience.

Further testing to compare the results obtained in our laboratories with the results obtained in their laboratories became complex. As noted above, our laboratory is not equipped to measure 2 x 2-ft mufflers and theirs was not equipped to measure 2 x 1-ft mufflers. We decided to attempt an investigation of the effect of modular size on noise reduction, since they had measured a 2 x 2-ft muffler and I had originally presented data on a 2 x 1-ft muffler. There was, therefore, a possibility of bias if significant "size" effects existed.

The 2 x 2-ft muffler tested in Company A's laboratories had a width at the minimum constriction of the air path of only 2½ in. while the apparent design value was 3½ in. It was agreed that two additional units with a 2½ in. minimum width should be constructed for test by us. One was to have a cross-section of 1 x 1-ft and one was to have a cross-section of 2 x 1-ft.

To assess the effect of variations in width at the minimum constriction, a second 2 x 1-ft muffler was to be constructed with the minimum width of 3½ in. The relevant data all taken by my noise reduction method are given in lines B.1 through B.4 of Table I and by the shaded area in Fig. 1.

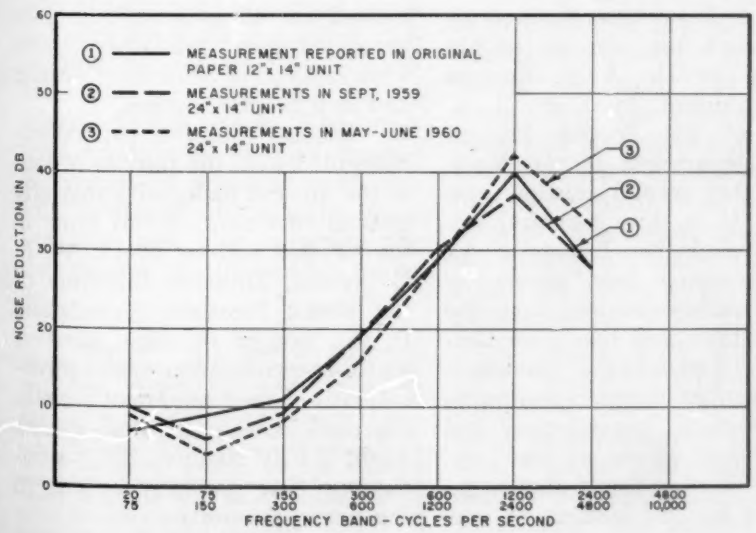
There is a small consistent difference between the noise reduction of the 1 x 1-ft muffler when compared with the 2 x 2-ft muffler. In general, the noise reduction of the larger muffler is greater. However, when we compare the noise reduction of the 1 x 2-ft muffler, we do not find that it lies consistently between the noise reduction of the other two mufflers as might be expected. There is a discernible trend towards an increase of noise reduction with size, but it is not established firmly on the basis of these tests.

Three of the mufflers which were retested had a width at the minimum constriction of about 2½ in., which is somewhat smaller than the apparent design goal of 3½ in. We therefore tested a muffler with a minimum width of 3½ in. to see if this parameter would influence the noise reduction methods. Comparing lines B.2 and B.4 we see there was a small difference noted. The muffler with the wider opening had a slightly lower noise reduction, about 1 db when averaged over frequency. Again no firm conclusion can be established

Table I Tests Summary

	Frequency Band—cps						
	20 75	75 150	150 300	300 600	600 1200	1200 2400	2400 4800
A. 1. 2 x 2 ft Co. A NR	8	11	9	17	32	47	31
2. 2 x 2 ft ND NR	8	8	10	14	32	40	31
3. Difference	0	3	−1	3	0	7	0
B. 1. 1 x 1 ft 2½ in.	6	8	9	14	28	37	23
2. 1 x 2 ft 2½ in.	10	8	8	13	27	37	30
3. 2 x 2 ft 2½ in.	8	8	10	14	32	40	31
4. 1 x 2 ft 3½ in.	10	6	8	12	20	36	26
C. 1. Difference B.3—B.1	2	0	1	0	4	3	8
2. Difference B.2—B.4	0	2	0	1	−1	1	4

Fig. 2 Noise reduction of two 4-ft long units (muffler C-3 in original paper)



from these experiments because the density of the glass fiber packing was varied (inadvertently) during these experiments, as well as the widths. However, the noise reduction does change slightly in the anticipated direction. The width of the minimum constriction of the mufflers represented by the data in curves 1 and 2 of Fig. 1 is not known.

When a representative of Company A brought mufflers A-4 and A-5 to us for test (September, 1959), he also brought Unit C-3 with a cross-section of 24 x 14 in. An attempt was made at that time to measure the 24 x 14-in. unit in our 24 x 12-in. concrete duct. The results (curve 2 in Fig. 2) oscillated above and below the original results (curve 1). The data in curve 1 were obtained from a 12 x 14-in. unit which exactly fitted a 12 x 14-in. duct.

Of special interest was the change in noise reduction at low frequencies. The first band increased and the second band decreased significantly. This behavior is not typical of any dissipative mufflers and it must be assumed that at the low frequencies the change in duct size affected the efficiency of the anechoic termination. At the time I attributed all of the differences to the unfortunate mismatch between duct size (12 x 24 in.) and muffler size (14 x 24 in.).

We attempted a retest of this 24 x 14-in. C-3 unit during the retests of the A units. The results obtained are given in curve 3. Again, we note the peculiar low frequency variations. The results are essentially the same as the results in our first attempt to measure this muffler, although the low frequencies decreased somewhat and the high frequencies increased a good deal. Again, the low frequency behavior suggests the termination is not as good as one would desire.

As in the case of the data shown in curves 1 and 2 in Fig. 1, the exact dimensions (minimum open area, density of glass fiber, etc.) of the original muffler tested are not known. Therefore it is not possible to define clearly how much of the variations in noise reduction result from variations in materials, variations in manufacture and experimental error. As before, the effects of total modular cross-section on noise reduction are not defined clearly. I would anticipate that one generally would find a slight increase in noise reduction at high frequencies with increasing module size.

SUMMARY AND CONCLUSIONS

A range of data has been obtained on two muffler units of different design. There seems to be a well-established difference in results corresponding to two different testing techniques. There remains, with a single testing technique, a range of noise reduction values. The author anticipated a smaller range, perhaps ± 1 db, based solely on experimental errors. A number of questions remains unsolved.

We have not been able to separate clearly variations caused by manufacturing tolerances from variations caused by changes in total cross-section. It may also be possible that experimental variations should be more than 1 or 2 db. A series of round robin tests, now being carried out under the auspices of American Standards Association S1-W-42, "Measurement of Sound Attenuation Devices in Ducts," may yield further insight to some of these problems.

Manufacturer A points out that they no longer manufacture Units A-1, A-2 and A-4. Manufacturer B also has developed a new line of mufflers. However, the important points illustrated by Figs. 2, 3, 4, 5 and 7 of my original paper, which are based partially on "old" mufflers, are not affected.

Air-Ionization Conference Explores Effects On Comfort and Health

At the International Conference on Ionization of the Air, arranged by the American Institute of Medical Climatology, October 16 and 17 at the Franklin Institute, Philadelphia, papers on the physiological and psychological effects of air-ionization were presented by 21 international authorities in the field at three technical sessions.

On the first day of the Conference, the first technical session, The Physics of Air Ions and the second, Physiological Effects of Air Ions, together, comprised a total of 17 addresses and included those of representatives of the engineering and medical professions, government agencies, private industry and scientists from the United States, Sweden, Russia and France.

One of the features of the first technical session was an address on Air Ion Measurement by ASHRAE Member J. C. Beckett, President, Palo Alto Engineering Company. A film, Ions In the Air, prepared by Dr. A. P. Krueger and introduced by Capt. J. L. Kinsey of the Special Projects Office, Department of the Navy, ended the second session. On October 17, a third technical session, Psychological Effects of Air Ions, presented four papers by three scientist-educators from the United States and one from Germany and a round-table discussion.

A number of papers dealt with experiments on physiological and psychological effects of ions. Dr. John L. Worden found that negative ions favored healing of cuts on the skins and recovery from shock in rabbits. Drs. A. J. Levine, M. Finkel, J. Handler, and W. I. Fishbein observed that patients treated for 45 minutes with ions generated from ultra-violet sources were relieved in 30 cases of asthma and in 9 bronchial cases.

Another group of papers was concerned with the physics of ions in the air and included a masterly general treatment of this topic in the banquet address by Dr. W. F. G. Swann, Director Emeritus of the Bartol Research Foundation. It was thought by some speakers that a re-examination of the physical basis of the experimental methods used in many of these studies might clarify some of the contradictions that presently appear in the reported results.

Solar Heat Gains

through three types of flat window glass

L. F. SCHUTRUM

Member ASHRAE

Much information has been gathered over the years on solar energy transfer through window glass and, more recently, the thermal evaluation of various shading devices. Glass coatings and films designed to limit solar transmittance and reflect or absorb infrared radiation have been in existence for some time, while laminated glass and plastics have made quite recent appearances in the window field. The purpose of most of these measures is to control or restrict visual radiation for lighting comfort, visible and infrared for thermal comfort, and near-ultraviolet for its adverse fading effects. The information presented in the following is concerned primarily with the total heat gain through the glass, but one must also recognize the importance of illumination, control of glare, the psychological desire of people to look at the out-of-doors, the temperature of the light and its effect on comfort, and architectural and aesthetic considerations.

The data set forth here were obtained at the ASHRAE Laboratory over a period of two-and-one-half years and are presented now to give some guidance to those who may be required to evaluate heat gains through these and similar fenestrations.

Any window glass, with or without film or coating, has certain characteristics for solar radiation. Sunlight falling on a glass will be reflected to the outside, absorbed by the glass, or transmitted to the indoors. The total solar energy falling on the glass is accounted for in these three ways in varying proportions, depending upon the characteristics of the glass and its coating. Thus, if a glass has a transmittance of 50% and absorptance of 30%, the reflectance must be 20% to total 100% of the solar energy falling on the glass.

Most window glasses have low reflectances for solar radiation, usually of the order of 5 to 10% at normal incidence. The solar reflectance of the plastics-coated glasses of the type coated in the field, which are transparent in nature, were found to be about the same as for uncoated glass. Similar but translucent plastics-coated glass which was tested at the ASHRAE Laboratory had higher solar reflectance, ranging from about 16 to 22%. Glasses with fused coating applied at the factory showed a range of reflectance of from 22 to 36%. With the fused coating facing indoors the reflectances are about 10 to 20%. Still another type, a laminated glass with an internal reflective coating, was tested. The reflectances of these laminates were found to range from about 20 to 70%, depending upon the reflectance of the coating.

Table I shows the transmittances and absorptances for solar radiation of unaltered window glass, coated glass, and laminated glass. Also shown are the multiplying factors which are applicable to various tables in the ASHRAE GUIDE for determining solar heat gains through windows for the August 1 design day. These data for common window glass and regular plate glass were taken from the GUIDE, but the transmittance value reported for the heat absorbing glass was increased slightly to be more typical of the heat absorbing glasses in use today.

The data reported for the coated and laminated glasses were obtained by two methods. Transmittance and reflectance measurements were made by means of an Eppley pyrheliometer under sunlight conditions. The transmittance for solar radiation was measured at normal incidence and the reflectance at near normal incidence angles. The absorptance of solar radiation was obtained by subtracting these two values from unity. The other method for getting engineering data was to utilize the ASHRAE Solar Calorimeter[1] for testing one sample of each of the transparent plastics-coated, fused coated and laminated glasses.

Only a limited number of samples was available, and the average transmittance and absorptance values given in the table are in round figures. One sample of the four translucent plastics-

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L. F. Schutrum was a Research Supervisor, ASHRAE Laboratory, until its closing. This paper was prepared for presentation at the ASHRAE Semiannual Meeting in St. Louis, Jan. 29-Feb. 1, 1962 and is a summary of data collected on this paper's subject at the Laboratory. Because of the Laboratory closing, not all of the desired tests in the program were completed. The author feels that, while the report is not as long or as complete as the usual research paper, the data presented should prove useful to design engineers.

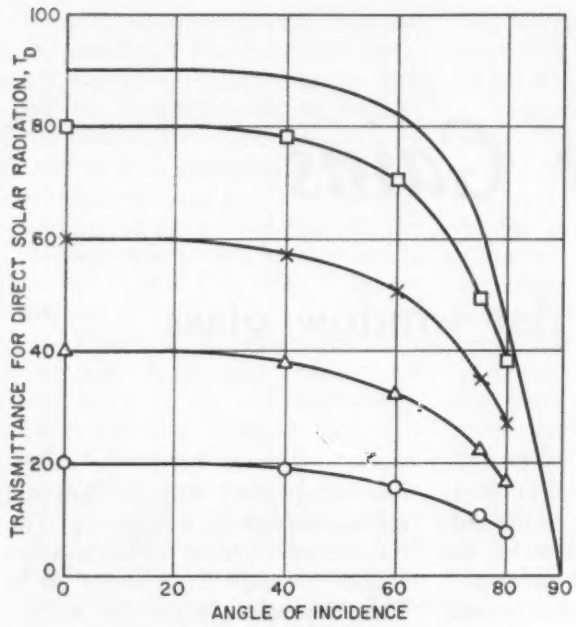

Fig. 1 Approximate variation of transmittance for direct solar radiation with angle of incidence

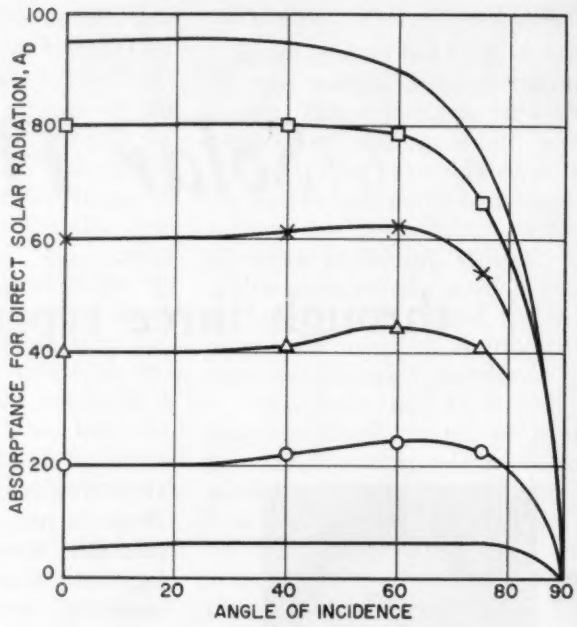

Fig. 2 Approximate variation of absorptance for direct solar radiation with angle of incidence

coated glasses proved inconsistent and was omitted from the average.

The fused coating applied to the regular plate glass was slightly different from that on the common and heat-absorbing glasses. The values entered in the table were estimated for the same type of coating as for the other two glasses.

The reflectance of the laminated glass and plastics depends upon the type of reflective film used in the sandwich. One sample not reported in the table had a solar reflectance of about 70%. The values in Table I for these laminates are reported to give the reader some idea of what has been done with laminates for controlling the solar heat gains.

The position of the coating on the outdoor or indoor side of the window was found to be important to the thermal performance only in the case of the fused coating. The values reported are for the coating on the outdoor side of the glass.

Table I Properties and Application Factors for Various Fenestrations

1	2	3	4	5	6	7	8
Window Glass	Modification	Samples Evaluated	Trans[a] T_{DN} (T_d)	Absorp[b] A_{DN} (A_d)	Multiplying Factor to Apply to GUIDE Table[c] 12	13	15
Common	None	—	0.87 (0.79)	0.05 (0.06)	1.0	1.0	0.0
Regular Plate	None	—	0.77 (0.69)	0.16 (0.16)	0.87	1.0	0.25
Heat[d] Absorbing	None	—	0.45 (0.43)	0.49 (0.47)	0.52	1.0	0.9
Common	Transparent[e] Plastics (Green, Blue, Grey)	5	0.45 (0.43)	0.5 (0.5)	0.5	1.0	0.9
Common	Translucent[e] Plastics (Green, Grey)	4	0.3 (0.3)	0.5 (0.5)	0.35	1.0	0.9
Common	Fused Coating[f] (outside)	3	0.4 (0.4)	0.3 (0.3)	0.45	1.0	0.5
Regular Plate	Fused Coating[f] (outside)	1	0.35[g] (0.35)	0.35 (0.35)	0.4	1.0	0.65
Heat Absorbing	Fused Coating[f] (outside)	3	0.25 (0.25)	0.5 (0.5)	0.3	1.0	0.9
Common	Laminated[h] Plastics	2	0.1 (0.1)	0.4 (0.4)	0.12	1.0	0.75
	Laminated[h] Plastics	1	0.3 (0.3)	0.5 (0.5)	0.35	1.0	0.9

[a] T_{DN} is direct normal solar transmittance. (T_d) is estimated diffuse transmittance for all but common and regular plate glasses.
[b] A_{DN} is absorptance for near normal incidence angle, and (A_d) estimated diffuse absorptance.
[c] ASHRAE GUIDE 1960, pp. 196 and 198.
[d] Heat absorbing glass in use today has slightly higher transmittance than recorded in ASHRAE GUIDE.
[e] Applied to glass after installation.
[f] Applied at factory.
[g] Observed transmittance was 0.4 with slightly different coating. 0.35 judged to be consistent with common window and heat absorbing glass.
[h] Reflective coating on inside of two sheets of glass (various types).

For plastics-coated and laminated glass, either side facing the outdoors gave about the same results.

APPLICATION FACTORS

To determine the heat gains through sunlit windows, the multiplying factors given in columns 6, 7 and 8 of Table I may be applied to the proper tables in the ASHRAE GUIDE. It is assumed that the variation in transmittance and absorptance is similar to that for common window glass, and this premise is supported partially by the tests made with the Solar Calorimeter. Furthermore, the emissivities of all the surfaces were found to be about the same as for ordinary glass, thus the combined outdoor and combined indoor coefficients of heat transfer should be about the same as for ordinary glass. For these reasons, the application factors should be acceptable for calculating solar heat gains. As an illustration of the GUIDE method for calculating solar heat gains through uncoated common window glass, select a west-facing window at 3:00 p.m., August 1, 40° N. Lat. The equation giving the total heat gain through the sunlit window is:

$$Q = (\text{Table 12 Value, GUIDE})\ (\text{Multiplying Factor}) + (\text{Table 13 Value, GUIDE}) \times (\text{Multiplying Factor}) + (\text{Table 15 Value, GUIDE})\ (\text{Multiplying Factor}) \quad (1)$$

where the multiplying factors can be found in columns 6, 7 and 8, respectively, of Table I.

The instantaneous heat gain for common window glass is 180 Btu/hr/sq ft from Table 12, p. 196, 1960 ASHRAE GUIDE. The multiplying factor is 1.0. From Table 13 of the GUIDE, the heat gain due to convection and radiation for the 95 F outdoor and 80 F indoor temperature is 21. The multiplying factor from column 7 of Table I of this report is again unity. The heat gain in Table 15 is 31 Btu/hr/sq ft and the multiplying factor from column 8 is zero. The total heat gain is then

$$Q = 180 \times (1.0) + 21 \times (1.0) + 31 \times (0) = 201 \text{ Btu/hr/sq ft} \quad (2)$$

In a similar manner, the multiplying factors of the coated and laminated plastics windows may be used to determine the heat gains.

A more general, but probably less accurate, method for determining heat gains is by the following formula[2,3]:

$$Q = K_D I_D + K_d I_d + U (t_o - t_i) \quad (3)$$

where

Q = instantaneous heat gain Btu/hr/sq ft of sunlit glass,

$$K_D = T_D + \frac{U}{h_o} A_D,$$

$$K_d = T_d + \frac{U}{h_o} A_d$$

I_D, I_d = direct and diffuse incident solar radiation, respectively, Btu/hr/sq ft

U = over-all heat transfer coefficient, air-to-air, Btu/hr/sq ft/F

t_o, t_i = outdoor and indoor air temperature, F

h_o = combined outdoor coefficient, Btu/hr/sq ft/F

T_D, T_d = transmittance for direct and diffuse solar radiation

A_D, A_d = absorptance for direct and diffuse solar radiation

Since U is of the order of unity and h_o usually is taken as 3 or 4, $\frac{U}{h_o}$ equals approximately 0.3. (This value is actually a variable but will be considered constant for the sake of simplicity.) For normal incidence direct solar radiation:

$K_{DN} = T_{DN} + 0.3\ A_{DN}$ where T_{DN} and A_{DN} can be found from Table I.

The transmittance and absorptance will vary with incidence angle approximately as shown in Figs. 1 and 2. The K_d value for diffuse solar energy is essentially constant for practical purposes and can be estimated from the data given in Table I.

Example: Solve the previous problem by means of Equation (3). First, find the incident direct and diffuse solar radiation. Table 6 of the 1960 ASHRAE GUIDE, p. 188, gives an altitude angle of 45.5°. Table 4, p. 187, for clear atmospheres, gives the direct normal radiation for an altitude of 45.5° as about 266.5 Btu/hr/sq ft and a diffuse radiation of about 35. The cosine of the incidence angle from Table 5 is 0.673 (this K is not to be confused with that of Eq. (3)) and the incident direct solar radiation $266.5 \times .673 = 179$. The incidence angle θ can be found from $\text{Cos}\ \theta = 0.673$ where $\theta = 47.7$ deg. From Fig. 1, T_D = about 0.84 for $T_{DN} = 0.87$, and from Fig. 2, $A_D = 0.06$ for $A_{DN} = 0.05$. The transmittance for diffuse radiation (T_d) is 0.79 and the absorptance (A_d) is 0.06 as found from columns 4 and 5 of Table I.

Thus $K_D = 0.84 + 0.3\ (0.06) = 0.86$
and $K_d = 0.79 + 0.3\ (0.06) = 0.81$

Equation (3) becomes

$$Q = 0.86\ (179) + 0.81\ (35) + 1.0\ (95 - 80) = 154 + 28 + 15 = 197 \text{ Btu/hr/sq ft}$$

where

$U = 1.0$, $t_o = 95$ F and $t_i = 80$ F

The heat gain previously calculated by the GUIDE method was 201 Btu/hr/sq ft.

If the window had been heat-absorbing glass, the first, or GUIDE, method of calculation would give 142.5 and the second, 136 Btu/hr/sq ft of sunlit glass.

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2. Solar Heat Gains Through Slat-Type Between-Glass Shading Devices by N. Ozisik and L. F. Schutrum, ASHRAE JOURNAL, Vol. 2, No. 9, Sept. 1960, p. 44.
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We state the case for

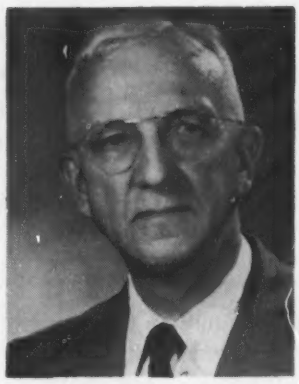
Ammonia as a Refrigerant

Why does anhydrous ammonia continue to be used in most commercial refrigerating and many air conditioning systems?

Why are so many new industrial refrigerating plants being designed and built to use ammonia as a refrigerant?

Why is it that this refrigerant receives so much unfavorable publicity?

To answer these questions in as few words as they are asked would often leave the sincere inquirer the choice of accepting a dogmatic reply or continuing his quest for knowledge. But the importance of true answers to these questions for refrigerating plant builders and operators justifies an answer that is neither brief nor dogmatic.



CLARENCE C. SMITH
Member ASHRAE

M. W. GARLAND
Member ASHRAE

INTERRELATED FACTORS

The choice of refrigerants has a basic effect on both cost of construction and operating costs. These in turn affect servicing and invested capital, as well as depreciation and production costs. Most people think of production costs as being affected primarily by labor and power charges – and they are right. But these in turn are dependent upon the equipment of the plant in question.

Let us take the last of our three questions first. A certain amount of unfavorable publicity seems unavoidable in a country enjoying freedom of the press. A healthy respect for occasionally-invoked libel laws does hold down improper references to individuals, especially when definite damages to the injured can be assessed.

But what can be done where the maligned party is a chemical composition? How can it defend itself? Who will defend it? A corporation is a body recognized by law and is entitled to sue and be sued – but a chemical composition enjoys no such privileges!

When publicity is given to reports of an accident, and made more sensational by blaming the most widely known refrigerant, ammonia, as the cause, what does the publisher have to fear if he has failed fully to report the truth? For example:

An explosion wrecked a large refrigerated warehouse and killed seven people in Richmond in January 1958; headlines in the public press featured the cause as an ammonia explosion.

Later, a $311,000 verdict based on contrary testimony was won by the owners and operators of the building and families of some of the victims. The verdict was sustained when it was appealed to a higher court. Natural gas leakage from city mains was determined to be the cause. But the papers never reported that the original articles had given the wrong cause.* No attempt was made to break down the prejudice that had been built up against ammonia.

Two years before, in Toledo, a similar blast caused by a leaking gas main had been featured by newspapers as an ammonia explosion.

In January of last year there was a fire in a cold storage warehouse at Faligsburg, Quebec, Canada, and reports cited ammonia as the refrigerant with rumor of some mechanical part of the refrigerating system as having failed and caused the fire.

The facts were that the plant was refrigerated by a halo-carbon. The fire did not originate in the compressor room, and the plant manager emphatically stated the fire did not originate with the refrigeration system.

Newspapers and public officials when reporting accidents, such as wrecks, explosions, mine disasters, etc., usually state that "the cause has not been determined" prior to official investigations. But in the case of the "refrigerant" accidents just mentioned, reporters assigned a cause before an official investigation and have blamed ammonia in a sensational manner.

Other publicity media are guilty of similar improper reporting. One of a series of rescue features on a TV show had one program based on fancied horrors

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Clarence C. Smith, retired, was associated with the H. C. Frick Company for many years. He is now a Consulting Engineer. M. W. Garland is Vice President, Engineering there. This is a somewhat condensed version of "Ammonia, the Slander-Resistant Refrigerant" as presented originally before the Baltimore Chapter of ASHRAE in early 1961 by Author Smith.

* The Atlantic Company vs City of Richmond in U. S. District Court, Eastern District of Virginia, Richmond Div.

caused by an ammonia leak when an earthquake was supposed to have struck an ice and cold storage plant. To anyone familiar with such installations the show was ludicrous. The type of technical competence directing it may be illustrated by their having men in an ice storage room make frantic efforts to save themselves from ammonia fumes by building a barricade of 300-lb. cakes of ice! Another instance was the earnest exhortation of one character telling another the dangers of ammonia fumes burning unprotected skin and to be sure there was no exposed surface on his arms between the top of his gloves and the bottom of his sleeves – while his ears and neck stuck out as they usually do beyond a regulation ammonia mask.

To a refrigerating engineer the ridiculous features come too fast to count; but to the uninformed public, ammonia was clearly the villain in the piece.

FOR AN IMPARTIAL VIEW

Some consulting engineers have a blanket policy of advising their clients against using ammonia as a refrigerant. In some such cases their organization may not include engineers capable of designing a safe, efficient ammonia plant; thus being unable impartially to compare and evaluate the advantages to the owner of different refrigerants as they affect plant design.

Many industries with long records of using refrigerating equipment have their own engineering staffs, or work with consulting engineers who do have a background of practical experience with ammonia as a refrigerant. They continue to design and supervise the installation of such plants, because of their knowledge of satisfactory results gained through practical experience.

Manufacturers of refrigerating machinery universally produce equipment for use with halo-carbon refrigerants. But the larger makers also continue making ammonia equipment. Some recently have expanded their operations into a wider coverage of the ammonia refrigerant field. They do so only because of the active demand for it from users of refrigerating plants.

Presently, no organization has sufficient interest in ammonia as a refrigerant to prompt a defense of it against the aspersions cast upon it.

So, the answer to the third question, as to why ammonia as a refrigerant receives so much unfavorable publicity, is that it has no organized defenders and the detractor is not concerned about his legal responsibility.

THE CASE FOR AMMONIA

Answers to the first two of our questions relate to the ability of ammonia to serve, economically and safely, the industries that use it properly as a refrigerant. What else could enable ammonia to resist and survive such an array of handicaps as outlined above?

And not only does it survive, but it is obtaining an increasing proportion of refrigerating work.

To many, that statement may be surprising, but the Department of the Census, "Facts for Industry," reference air conditioning and refrigerating equipment 1959, released September 24, 1960, shows that in sizes above 20 hp in 1959 all reciprocating compressor sales increased over 1958. Ammonia compressor sales increased 19%, while halo-carbon compressor sales rose but 6%. If one year's figures might not be considered long enough to show a trend, compare three years' – 1957 to 1959 – here the totals show the number of ammonia compressors in this class rose 27% and the halo-carbons but 12%.

The cumulative figures show ammonia compressor sales to have risen 21.5% and the halo-carbons but 2.7% in the same class. So you see, ammonia is more than maintaining its position in the refrigeration industry.

Three of the larger brewing companies each built a new brewery in the Southeast in the last two years. Each of them uses ammonia as a refrigerant; some of the companies previously have built plants using other refrigerants.

Cost conscious meat packers, whose operations are well known to be highly competitive, are using ammonia as a refrigerant almost exclusively. The same is true of cold storage warehouses.

The newest and one of the largest ice cream and dairy plants constructed in Florida uses ammonia.

Processors of frozen foods from Maine to California and from Florida to Alaska are almost exclusively users of ammonia for refrigerating purposes. This is true of distributors of frozen foods, too.

Big grocery chains continue to install ammonia systems for their warehouses, using it economically and satisfactorily to refrigerate a wide variety of fruits, vegetables, dairy and meat products.

But the use of ammonia is not confined to large plants. Hundreds of skating rinks throughout the USA and Canada use it. Whole fleets of fishing and shrimp boats use it. Large and small chemical, rubber processing plants and oil refineries continue to install ammonia refrigerated plants.

Many large and small air conditioning installations use ammonia as a primary refrigerant and continue to do so safely and economically. Two of the larger buildings in Tulsa, Okla., have a 1,000-ton common system that has used this refrigerant for more than 22 years, with low maintenance and high economy.

Granted that ammonia is widely used, that its position in the industry is not only being maintained but increased, we still have to answer the question – why does it face so much opposition?

Many objectors to ammonia

One of the deservedly controversial topics in the fields of refrigeration and air conditioning is the proper place or acceptance of ammonia as a refrigerant. "More sinned against than sinning" is the verdict of these authors and, with light of the hard facts of performance and economics, their contention surely deserves a fair-minded reappraisal. Sooner or later, it is bound to happen.

as a refrigerant claim concern with safety. Now "Health" and "Safety" are two words more used by self-serving groups than any other two in the language.

There is no such thing as absolute safety and the term, being relative, when applied to refrigerating plants means proper design for relatively safe installations, regardless of the refrigerant used. There is no way to prevent people being injured or killed due to ignorance or carelessness of themselves or of other people no matter what the specific problem. Putting a blow torch to a refrigerant drum containing a halo-carbon refrigerant would be folly, too, had it contained ammonia.

A leak in a system spraying out oil vapor with the refrigerant, in the presence of a spark, can cause an explosion regardless of the refrigerant used. Indeed, those systems containing refrigerants miscible with oil may be more dangerous in this respect than ammonia systems which circulate less oil in proportion.

The breakdown or disassociation of the elements comprising any refrigerant, halogenated or other, is a highly technical and complicated matter; full consideration of it would involve a discussion much longer than permissible in this article. It is sufficient to note that such instances have been extremely rare over a period of many years' operation, which, with ammonia, extend back nearly a century. Ammonia compressors built 65 years ago are still in operation.

JUST ANOTHER UTILITY

Observance of the codes governing refrigerating plant installation, and modern methods of testing and development by reputable manufacturers of ammonia equipment, have made this equipment as dependable and safe to use as steam, natural gas or electricity. The inclusion in plant design of safety relief valves, cutouts actuated by thermal as well as pressure elements, provision for oil recovery, compressor room ventilation, effective foul gas purging systems, proper operating compression ratios and prevention of air and water infiltration are requirements of any refrigerating system regardless of refrigerant used.

Important reasons for ammonia's continued and expanding use as a refrigerant are lower first cost of the refrigerant itself; easier and quicker location of leaks, resulting in lower renewal requirements and costs; wider availability, especially in foreign fields, which take our exports of refrigerating machinery.

The halo-carbons weigh much more per cu ft than ammonia. Their cost per pound varies from 2.5 to 5 times more than ammonia, the weight per cu ft is 2 to 4 times more, so this may be a considerable item running into thousands of dollars in some installations. In some cases with odorless halo-carbons, leaks are not detected until the entire charge is lost, or product temperatures rise to damaging heights. Spoilage costs add to the expense of recharging the plant with high-priced refrigerant. The pungent odor of ammonia reduces this hazard.

In some cases ammonia refrigerated plants are lower in first cost and in some cases halo-carbon refrigerated plants are cheaper to install. There is no rule of thumb that can be applied universally. Each installation should have its requirements analyzed in an unprejudiced way.

The design of long suction lines, especially when inter-connected with multiple low pressure evaporators, creates the problem, with halo-carbon refrigerants, of maintaining sufficiently high gas velocities through vertical risers and traps to return oil, without sacrificing pressure drops that penalize horsepower consumption per ton of refrigeration produced. This is not a problem with ammonia.

In an effort to reduce the penalties imposed by long suction lines and scattered evaporators, a multiplicity of halo-carbon compressors may be installed. This increases the initial cost as well as maintenance charges and operating power consumption.

In one meat packing plant, 10 halo-carbon compressors totaling 80 hp and their attached equipment were removed and two 30-hp ammonia compressors with high and low sides were installed, with most satisfactory results. Many other examples could be listed. Examples of ammonia refrigerated plants being replaced can also be made, but it is far from a one-way street, as some propaganda would infer.

To escape the disadvantages of multiple halo-carbon compressor installations, a central refrigerating plant circulating water or brine may be considered. Besides the losses inherent in any secondary refrigerating system, such as temperature splits, there is the cost of pumping and removal of the heat which pumping generates – there are approximately double refrigerant line losses. Where more than one temperature is required, in these cases, there is the need to operate at the lowest, most uneconomical pressure level for the total load.

Ammonia, with its relative freedom from oil, permits a much better heat transfer through surfaces which cannot be designed for high refrigerant velocities. Shell vessels such as direct expansion ice cream freezers and fragmentary ice manufacturing vessels are examples. The physical fact that ammonia is lighter than oil readily permits drainage of oil from the bottom of refrigerant containers, while the ammonia floats on top. The circulation of quantities of oil in a halo-carbon refrigerating plant makes parallel machine operation difficult because crank case oil levels in such cases are hard to maintain. This is no problem with ammonia compressors.

Great stress sometimes is laid on the fact that the theoretical horsepower requirements per ton of refrigeration vary but slightly regardless of the refrigerant used. This is true if the compressor only is considered. There is, however, a wide difference in power consumption of the whole refrigeration plant when using different refrigerants.

This is due to the difference in weight flow requirements and the resultant pressure drop being greater with halo-carbons than with ammonia. Also due to line velocity requirements mentioned before; difficulties in connection with refrigeration flow controls; and lower refrigerant temperature requirements due to oil coating the evaporators.

With halo-carbon installations where load variations are of suffi-

cient magnitude and duration to require more than one compressor for efficient operation, separate low side refrigerant circuits often are required and it may be necessary to install separate condensing circuits as well. This is more expensive and complicated than a similar ammonia plant, to install as well as to operate.

In plants where multiple suction pressures and varying loads occur, ammonia systems have a flexibility and simplicity not possessed by halo-carbon systems. This is especially true in connection with labor and power saving controls.

These are some of the answers to the first two of our questions as to why ammonia continues to be used in refrigerating plants today and why many new plants are being built for use with this refrigerant.

The wide extent of the use of halo-carbon refrigerants and ammonia demonstrates that there is a field for both. The choice, in any case, should not be an arbitrary one.

Plans Taking Shape for

Semiannual Meeting at St. Louis

January 29 — February 1, 1962

As tentatively planned, the four-day ASHRAE Semiannual Meeting, January 29-February 1, 1962, will contain a varied and interesting series of technical, exploratory, business-session and social events.

Four Technical Sessions will be held during the Meeting, at which a total of 21 papers will be read.

The First Technical Session, on January 29, will be composed of papers on the Operating Economy of Air-Cooled Refrigerant Condensing, Cross-Flow Cooling Tower Analysis and Design, Acid Removal by Various Desiccants and the Selection of Hot Junction Temperatures for Minimum Fin Area Thermoelectric Refrigerators. The Second Session (January 30) will introduce Controlled Atmosphere Apple Storage, Field and Laboratory Performance of Air Cleaners, Operating Principles of Non-Ionizing Electrostatic Air Filters and Fabric Filter Cleaning by Intermittent Reverse Air Pulse.

On January 31, the Third Technical Session will present Improved Methods for the Calculation of Design Temperature, Radiant Air System Design and Performance, Solar Heat Gains Through Plain, Coated and Laminated Flat Window Glass and Energy Sources and Requirements for Residential Heating. The concluding Technical Session (February 1) will include An Instrument for the Measurement of the Humidity of Air, which will be presented by title only, Electric Analog Computer Solution of Combined Heat and Mass Transfer, Heat and Mass Transfer in Dehumidifying Surface Coils and An Analog Evaluation of Methods for Controlling Solar Heat Gains Through Windows.

Important, if less formally conducted, will be six Symposiums and four Forums. The Industrial Ventilation Symposium will be made up of two parts: Make-up Air and Air Distribution (January 29) and Noise Generation (January 31). The other Symposiums will be: Combined Heating and Cooling Systems (January 29), Domestic Refrigeration Engineering (January 30), Steel Boiler Ratings (January 30), Room Air Conditioners (January 31) and Food Storage (February 1).

Advance committee meetings are planned for January 27-28, as well as Advance Registration and a Reception and Tea, with a tour of St. Louis in private cars, on January 28. Social events forecast for the Meeting comprise a Welcome Luncheon, Climatron and Hot Punch Party and visit to the National Council of State Garden Clubs, Dinner and Floor Show on January 29; Life Members' Luncheon and Past Presidents' Dinner on January 30; Ladies' Continental Breakfast and Play Review, Dutch Treat Cocktail Party, Entertainment and Dancing on January 31.

A complete program of the St. Louis events, with abstracts of technical papers to be presented, will appear in the January issue of the JOURNAL. Preprints of the Technical Session papers will be available at the Meeting. Some of these papers will have appeared in the JOURNAL before Meeting time.

ASHRAE NATIONAL MEETINGS AHEAD

1962	1963	1964
Jan. 29-Feb. 1 Semiannual St. Louis, Mo.	Feb. 1-14 Semiannual New York, N. Y.	January 27-29 Semiannual New Orleans, La.
June 25-27 69th Annual Miami Beach, Fla.	June 24-26 70th Annual Milwaukee, Wisc.	June 29-July 1 71st Annual Cleveland, Ohio

What ASHRAE Regions and Chapters are doing

As the new season got underway, Chapter meetings covered such diverse topics as central heat pumps, boiler ratings, ventilation of fall-out shelters, air sanitation and high velocity duct systems.

SOUTH FLORIDA . . . Speaking on September 12 was James N. Owens, Chief Engineer, Dade County Board of Public Instruction. Topic of his discussion was "School Heating and Air Conditioning." with special emphasis placed on Dade County schools. Conclusion resulting from studies made was that air conditioned schools could be built with less cost than conventional schools, because of compactness of structures and smaller glass areas.

Scheduled for October 10 were four speakers on humidity: R. A. Nance of BBB Engineering Service, covering infiltration; D. L. Howard of Trane Company, effect of infiltration and ventilation on space relative humidity and methods of reducing humidity by conditioning equipment; E. H. Morris of Parsons and Morris, humidity as it affects comfort, odor, mold and condensation; and John Lotz III of Industrial Insulators, Inc., mistakes in application of insulation and vapor barriers.

BOSTON . . . Professor Holt of MIT held a group participation seminar on boiler ratings at the October 17 meeting. Guest speaker B. Graham, Director of Research for Buffalo Forge Company, discussed sound as applied to fan systems. Various aspects of sound measurement were covered.

who's doing what . . . New Life Members are Norman J. H. Shaw, Carroll R. Swaney, Robert Wise, Donald J. Adams and Alfred J. Ferretti.

HOUSTON . . . History of the Texas Chapters was outlined at the September 26 meeting by Presidential Member R. F. Taylor, beginning with formation of the first Chapter at College Station in 1939. In that same year Texas was divided into two sections, the North and South Texas Chapters. In 1946 a Southwest Chapter was organized in San Antonio. There are now eight Texas Chapters: Alamo, Austin, Dallas, East Texas, El Paso, Fort Worth, Houston and West Texas.

Speaking next was A. F. Barnes, who was Chapter President in 1944-45. Covering "Heating Yesterday and Today", he advised that in the period following World War I, cast iron radiation was the standard, with no automatic control and no gas available for firing boilers. Heating systems consisted of the hot air gravity type, with single-pipe steam systems in general usage. Fan coil units gradually became more common and by 1926 pneumatic control systems made their appearance. Also introduced in the late 1920s were blow-through unit heaters, natural gas and convection. Packaged boilers, baseboard radiation and use of hot water rather than steam for heating were initiated after World War II.

Harold Broadwell, 1951 President of Houston Chapter, spoke on "Refrigeration and Low Temperatures", beginning with provision of man-made ice in the 1880s by means of plate-type ice plants. Advances in the field mentioned include blast freezing, refrigerated transportation and cryogenics. Outlining techniques utilized to reach extreme low temperatures, speaker Broadwell stated that in the −30 to −40 F range, two-stage compression normally is used; from −70 to −80 F three stages are utilized, usually two rotary and one centrifugal; for extremely low temperatures, multiple stage and cascade circuits are used.

Final speaker of the evening was Arthur Chase, President of Houston Chapter in 1943, who detailed "Early Air Conditioning in the Southwest", beginning with its inception for multi-story buildings in 1928. By 1930, air conditioning had evolved to the point where larger sized duct work with relatively large cfm capacities was being used. Automatic controls were unheard of, and various schemes for mechanical' damper arrangements were devised to provide unitized control for office spaces.

BRITISH COLUMBIA . . . First meeting of the new season was held on October 11 and featured a discussion by Chapter President D. B. Leaney of his trip to England to attend the meeting of the International Institute of Refrigeration. He reported on various papers presented at the meeting, especially those

CHAPTER MEETING DATES

	Dec.	Jan.	Feb.
Akron	12	9	13
Alamo	—	—	—
Arkansas	—	—	—
Atlanta	11	8	12
Austin	15	18	15
Baltimore	7	4	14
Baton Rouge	20	17	21
Boston	21	16	20
British Columbia	—	10	14
Central Arizona	4	4	5
Central Indiana	12	9	13
Central Michigan	12	9	13
Central New York	13	10	3
Central Oklahoma	—	—	—
Central Pennsylvania	—	—	—
Cincinnati	—	—	—
Cleveland	11	8	15
Columbus	18	15	19
Dallas	18	15	19
Dayton	12	9	—
East Texas	26	23	27
El Paso	—	—	—
Evansville	5	9	6
Florida West Coast	8	16	20
Fort Worth	—	—	—
Golden Gate	7	4	10
Hampton Roads	12	23	20
Houston	26	23	27
Illinois	11	8	12
Illinois-Iowa	—	15	19
Inland Empire	11	8	12
Iowa	17	15	19
Jacksonville	5	2	6
Johnstown	—	—	—
Kansas City	4	2	5
La Ville de Quebec	—	—	—
Long Island	11	8	12
Louisville	9	10	14
Manitoba	—	—	—
Memphis	—	—	—
Michigan	11	15	19
Middle Tennessee	12	9	13

concerning hospital air conditioning. Research Chairman Michael Lambert mentioned that at the next meeting he will report on the work his committee has done with regard to British Columbia summer and winter design data.

"Some Legal Considerations of Our Industry" was the topic considered by a panel comprised of Joseph Moss of Prudential Insurance Company, A. Grimrod of A. R. Grimwood, K. Nesbitt of Canadian Surety Company, S. Welsh of Fred Welsh & Son and G. Lyall of Bourne, Lyall, Shier and Davenport. Aspects covered were financing, bonding, loaning and general and mechanical contracting.

who's doing what . . . H. R. Cherewick is the recipient of the Chapter scholarship.

OREGON . . . Speaking at the September 14 meeting on the general topic of "Central Heat Pumps", R. C. Chewning detailed heat pump feasibility studies, load calculation, operating characteristics and control circuits and Omer Jacobson covered the air-to-water heat pump installation at the Mabel Rush Elementary School.

who's doing what . . . C. J. Bell has succeeded W. S. Cooper as a member of the Board of Governors.

Using slides throughout his address, D. L. Benz of Benz Company spoke at the October 12 meeting on "Air Filtration". Various facets discussed include a description of types of atmospheric dust encountered, details of three major test methods (weight, dust-spot and dust fed) and features of three types of filters: impingement, strainer and electronic.

DALLAS . . . Subject of the September 18 meeting was Civil Defense. Speaking were Weeden B. Nichols, Col. John W. Mayo (U. S. Army Retired), Dr. Jack Cromer and architect Delbert Westfall. Each presented his view on the problems of nuclear defense and solutions for survival in event of atomic attack.

Scheduled for October was a joint meeting with Fort Worth Chapter, featuring a talk by F. M. Johnson on air sanitation. (See FORT WORTH)

AKRON . . . Space flight was covered at the September meeting by guest speaker Derrel C. Romich, an aerophysicist for Goodyear Aircraft. Beginning with an explanation of conditions in and outside the atmosphere, he then detailed early rocket development, followed by discussion of present and planned space vehicles.

Among items discussed at the October 10 meeting were designation of November 2 as the next meeting date to coincide with presentation of the Charter to this newly-organized Chapter and attendance of members at the Regional Meeting in Columbus, October 27-28.

Speaker of the evening was Gerald McIlhenny of Ohio Edison Company, who detailed "Electric Heating". Covered were electric-powered baseboard radiation, duct-mounted heaters, immersion heaters and heat pump air conditioning.

SOUTHERN CALIFORNIA . . . Problems encountered in design of "White Rooms" at Autonetics Div of North American Aviation were discussed at the October 9 meeting by W. R. Jerome and David Mattox. White rooms must have a dust count below ten five-micron particles per cu in.

who's doing what . . . Bernhard Goerg and Siegfried Ruppright are new Life Members.

WESTERN MICHIGAN . . . In attendance at the October 9 Past-Presidents' Night were the following past Presidents of this Chapter: R. J. Arend, S. R. Curtis, R. L. Eichhorn, F. Harbin, H. R. Limbacher, H. Johnson, L. G. Miller, F. Murray, C. Pesterfield, F. G. Peck, S. N. Paganelli, D. J. Renwick and B. J. Walter.

Joseph C. Sloane, owner of Sloane Heating and Air Conditioning Company, spoke on "Application of Heat Pumps in Michigan". He indicated that the air-to-air heat pump has many practical advantages and when installed with auxiliary heating coils can be operated in areas where outside temperatures drop as low as −20 F. Several installations were described.

who's doing what . . . D. J. Renwick will be Chapter Delegate at the Regional Meeting.

MONTREAL . . . Scheduled for September 18 was a meeting on "What the Architect Expects of the Mechanical Engineer", with R. T. Affleck of Affleck, Desbarats, Dimakopoulos, Lebensold, Michaud, Sise, Architects, as guest speaker. Stressing the importance of teamwork, he stated that a major problem of communication exists in the area of basic conceptual design, with the engineer often ruling himself out of this phase of the work, believing that this is the concern only of the architect. Speaker Affleck contends that cooperation at this stage is of utmost importance, in view of the significance of mechanical, electrical and structural work in a total project.

	Dec.	Jan.	Feb.
Minnesota	11	8	12
Mississippi	15	22	26
Mobile	26	30	27
Montreal	18	15	19
National Capital	13	10	14
Nebraska	—	—	—
New Mexico	—	—	—
New Orleans	12	16	20
New York	—	23	27
Niagara Frontier	—	—	—
Niagara Peninsula	8	9	13
North Alabama	7	4	1
North Jersey	7	4	1
North Piedmont	—	—	—
Northeastern New York	11	15	19
Northeastern Oklahoma	18	15	19
Northern Alberta	—	18	24
Northern Connecticut	—	11	8
Northern Ohio	—	—	—
Ontario	4	4	5
Oregon	14	11	3
Ottawa Valley	—	—	—
Panama & Canal Zone	—	—	—
Philadelphia	14	—	8
Pittsburgh	—	15	19
Puget Sound	12	10	14
Rhode Island	13	10	14
Richmond	—	8	5
Rochester	6	10	17
Rocky Mountain	6	3	7
Sacramento Valley	—	—	—
St. Louis	—	—	—
San Diego	—	—	—
San Joaquin	5	2	6
Savannah	—	—	—
Shreveport	—	—	—
South Carolina	—	15	19
South Dakota	—	—	—
South Florida	12	10	14
South Piedmont	19	16	26
Southern Alberta	—	—	—
Southern California	11	8	12
Southern Connecticut	—	—	—
Toledo	—	—	—
Tucson	5	9	6
Utah	22	19	16
West Texas	—	—	—
Western Massachusetts	13	18	15
Western Michigan	11	8	12
Wichita	—	—	—
Wisconsin	—	—	—

Quantity surveying is an organized method or means of taking quantities from architects' or engineers' plans and specifications, for the use of contractors' estimating work, to indicate if the actual cost will be above or below the estimated quantity price. Covering this topic at the October 16 meeting was R. R. Firmin of G. A. Hanscomb Partnership. Included in such a survey are the amount of material, how and where it is to be set and deviations from the specifications in the equipment supplied. It is also the duty of the quantity surveyor to follow the job in its design stages, recommending changes that will keep the job within its budget price.

EAST TEXAS . . . On October 16 a meeting of the Board of Governors was held, to discuss arrangements for the October 24 Charter Meeting, reported on by Travis McLane, Warren Spofford and W. A. Arzberger. Other matters covered were Chapter meeting dates and proposed programs for the coming year.

FLORIDA WEST COAST . . . Regional Director J. G. Woodroof spoke at the October 17 meeting on "Freeze Drying and Other Methods of Food Preservation".

PITTSBURGH . . . Included in the talk of September speaker Bart Small of Aluminum Company of America, "Spot Cooling and Heating," were many problems of heating, cooling and ventilation in industrial plants. Various applications requiring spot heating and cooling were described, and installations were shown by means of slides.

NEBRASKA . . . E. Tidd of Bell & Gossett Company spoke at the September 12 meeting on his field experiences in correcting heating problems.

WISCONSIN . . . Chapter Past-President Robert D. Rodwell, Consulting Engineer, spoke at the September 18 meeting on the importance of a small businessman having an acquaintance with the financial aspects and requirements of his business.

Present at the October 10 meeting were mechanical engineering students from the University of Wisconsin, candidates for the Gustus L. Larson Award, presented annually in memory of Professor G. L. Larson, past Chairman of the Mechanical Engineering Department at the University. Also in attendance were Professor Del Nelson of the Mechanical Engineering Department and Robert S. Burdick, 1961-62 recipient of the award.

Wing Commander Jaffre Woolfendon of the Royal Canadian Air Force spoke on North American air defense. Assisting him were Captain Scarsborough, Public Relations Officer at Truax Field and E Stefani of Radio Corporation of America.

NORTHEASTERN OKLAHOMA . . . Measuring and control of sound level were discussed at the September 18 meeting by J. W.Spradling of Carnes Corporation. Emphasis was placed on the relationship of the noise level of air moving equipment to the type and size of the structure being conditioned.

On October 16 R. H. Jacobs of Minneapolis-Honeywell Regulator Company spoke on "Electronic Air Cleaners", with respect to their effect on air conditioned environments.

who's doing what . . . Jack Langston, a student at Tulsa University, was presented with the Chapter's Meinholtz Scholarship Award.

NORTHERN CONNECTICUT . . . Causes of hermetic burnout were detailed at the September 14 meeting by T. Atwood. Main points of his discussion were testing procedures and limits on contamination.

A tour was made of Hamilton Standard on October 12, with emphasis placed on air conditioning and environmental controls. Speakers for the occasion were H. A. Larkens, J. F. Raye and Anthony Calderone.

JACKSONVILLE . . . Progress being made in various sections of the country in school air conditioning was detailed at the September 5 meeting by Chapter members D. Langford and D. Russell.

CENTRAL ARIZONA . . . Presidential Member D. D. Wile spoke to members attending the October meeting about the United Engineering Center, prior to beginning the evening's main topic, "Modern Trends in Commercial and Industrial Refrigeration". Slides showed application in packing plants, commercial storage and industrial plants.

NEW YORK . . . Efficient winter operation of air-cooled condensers requires controls to cope with characteristic winter problems of low head and receiver pressures during the on cycle and low evaporator pressures during the off cycle, as well as problems of refrigerant migration. Discussing the origin of these problems and analyzing the various methods of solving them was October speaker Dan Kramer of Kramer Trenton Company.

Covering the problem of maintaining adequate head pressure, he detailed several methods, all of which essentially operate by reducing the effective surface of the condenser so that a sufficiently high head pressure is maintained to drive the liquid through the expansion valve. Various schemes for control of refrigerant migration during shut-down were reviewed from the aspect of migration occuring whenever the condenser surface is colder than the evaporator surface.

LOUISVILLE . . . Detailing design of fall-out shelters, October speakers were Col. Leon J. Reed (U.S. Army Retired) and Arthur F. Hubbard, American Air Filter Company. Speaker Hubbard took issue with the Office of Civil Defense Mobilization recommendations on what constitutes adequate ventilation. Ten rather than three to five cfm per person was the minimum he suggested. Utilizing typical heat balance calculations, ten cfm per person is needed to maintain a 70% R.H. level for a 90 F ambient.

Another point of contention was the use of a trap in the ventilating pipe to filter out radioactive dust, suggested by the OCDM. Mr. Hubbard stated

that furnace-type filters would be more effective. What is really needed for shelter ventilation, he said, is a filter that will protect against chemical and bacteriological contaminants as well as airborne radioactive dust. Such a filter for broad scale use is still in the development stage.

FORT WORTH . . . Regional Director W. J. Collins, Jr., spoke at the joint meeting with Dallas Chapter on October 16, discussing National activities and organization of a new Chapter in Tyler, Texas.

Speaker of the evening was F. M. Johnson, whose talk on air sanitation, covering purification and dehumidification, was illustrated with slides.

LONG ISLAND . . . Meeting on October 9, members heard Victor Stampfl of the Allen Bradley Company speak on reduced voltage starters, electrical components and their needs.

MIDDLE TENNESSEE . . . September speaker was Lee Stratton, Sales Engineer for Minneapolis-Honeywell Regulator Company, who covered "Flame Detection" as applied to gas and oil burner controls.

"Water Treatment and Conditioning as Applied to Air Conditioning Systems" was the subject of October speaker F. S. Hogdson, Assistant Chief Engineer, Metropolitan Refining Company.

ILLINOIS-IOWA . . . First meeting of the year was held September 18 at the Caterpillar Tractor Company, Industrial Engine Plant. Members in attendance toured the manufacturing area and were shown heating and ventilating equipment.

"Natural Gas Engines for Air Conditioning and Heat Pumps" were discussed by Jack Gill. To be acceptable, a gas engine, he stated, must have economic advantages, automatic operation, be reliable and be compatible to the equipment driven.

Speaking on October 16 was R. E. Kittoe of Meters and Controls, who demonstrated Flame Safeguard equipment.

SAINT LOUIS . . . Condenser operating problems in various outdoor ambients were discussed by October speaker Dan Kramer of Kramer Trenton Company. Low ambient on cycle problems pointed out were low head pressure and condenser logging. Also discussed was an important off cycle problem, refrigerant migration through the condenser valves. Optional control systems, using either condenser air flow or refrigerant control, were shown. In the opinion of the speaker, that system which protected against all problems mentioned consisted of a holdback at the receiver inlet, bypass valve between the discharge and liquid lines and a check valve in the liquid line.

Presenting a financial grant to the Department of Air Conditioning, Heating and Refrigeration of Hudson Valley Community College, Northeastern New York Chapter President B. E. Mullen makes the award to Department Head Paul Goliber as Otto B. Guenther, President of the College, looks on.

ATLANTA . . . Mechanical design of the new Atlanta airport was discussed by September speaker Robert Dean of Robert and Company. Outline drawings were utilized to indicate the extent and arrangement of the facility. Following the presentation, members were conducted on a tour of the airp[illegible].

Regional Director J. G. Woodroof and J. Kelly spoke October 9 on the Region IV meeting in Jacksonville October 18, 19 and 20. "Make-up Air Systems" was covered by David Webster of Reznor Manufacturing Company.

SAN DIEGO . . . Program topic for the September 12 meeting was "Air Conditioning Systems on Launching Pads," presented by D. Suchner of General Dynamics/Astronautics.

Meeting jointly with the local chapter of Illuminating Engineering Society, members attending the October 5 meeting heard W. S. Fisher of General Electric Company speak on "Electrical Space Conditioning", which covered integration of lighting, heating and cooling in modern building design.

ROCKY MOUNTAIN . . . Gerald H. Phipps, President of Gerald H. Phipps, Inc., spoke on October 4. Subject of his discussion was "What's Wrong With the Construction Business?"

ILLINOIS . . . Charles N. Rink of Industrial Acoustics Company spoke at the October 9 meeting on "Noise Control in Air Conditioning Systems". After briefly covering definitions and indicating how to separate the mechanical vibration problem from that of acoustical vibration, he spoke on solution of noise problems in heating, ventilating and air conditioning systems. Slides and charts emphasized various aspects of his talk.

SHREVEPORT . . . Charles R. Hays, Principal Meteorological Assistant at the U. S. Weather Bureau Station, spoke at the September 21 meeting on severe storm warning procedures and methods.

Performance of the three-pipe drilled and hot water air conditioning system was compared with other systems by October speaker E. Bruce Appling of York Corporation. Performance was illustrated graphically and cost comparisons were made.

MINNESOTA . . . Fall season began for this Chapter on October 9 with a talk on "Ionizing Radiation", presented by Dr. Warren R. Lawson of the Minnesota Department of Health. In dealing with the ventilation requirements for fallout shelters, Dr.

Lawson pointed out that dust is the basic carrier for the particulate matter which could create a radiation hazard. Filtering out this dust is a basic requirement for a shelter.

SOUTH CAROLINA . . . Ultra low temperature refrigeration at the University of Georgia, an installation ranging to −60 F, was discussed on October 16 by R. G. Hanna of Mangan & Hanna.

MICHIGAN . . . "Aerospace Activities at Cape Canaveral" was planned for discussion at the September 18 meeting, with Major M. E. Griffith, USAF, as guest speaker. To be included in the presentation were history of the Atlantic Missile Range, activities of the Air Force Missile Test Center and down-range stations.

MOBILE . . . Opening the season on September 28, 1961, was to be a discussion of "Centrifugal Pumps, Their Design and Application" by J. H. Wright, Application Engineer, Gould Pumps, Inc.

ARKANSAS . . . Members of a panel which discussed three and four-pipe versus two-pipe chilled-hot water systems at the October 17 meeting were R. E. Blaylock, Blaylock & Associates; T. W. Turner, Medlin Plumbing and Heating Company; P. Duff, Powers Regulator Company; and A. Bullard, Allen Bullard Company.

JOHNSTOWN . . . Defining high velocity air as that which is moved through ducts at 4500 to 6000 fpm, October speaker C. D. Jacobs of Anemostat Corporation discussed "High Velocity Air Distribution and Its Application." Major advantages cited were: 100% outside air can be used for cooling, design is simplified, less equipment to maintain, greater flexibility of control in mid-season operation and smaller ducts provide savings in space.

KANSAS CITY . . . Reports were made at the October 2 meeting by the following committee chairmen: R. S. Beeson, Public Relations; R. J. Walsh, Btu Breeze; Charles Shonfelt, Special Events; Lee Vogel, Reception and Welfare; P. E. Chamberlin, Vocational Guidance; and R. J. Ruth, Audit and Budget.

Schools should be air conditioned the year round for better education, stated guest speaker Z. A. Marsh of Minneapolis-Honeywell Regulator Company. Economy may be achieved by building schools square, round or as hexagons, and by having fewer windows and more wall space. Slides and charts illustrated his talk.

NIAGARA PENINSULA... Using slides, Walter Stethen of S. A. Armstrong spoke at the October 10 meeting on primary and secondary circuits for hot and chilled water systems.

TUCSON . . . Harold Derus of Waukesha Motor Company, speaking October 3 on "Engine-Driven Air Conditioning Equipment", discussed adaption of the gas engine for driving refrigeration equipment on railroad passenger service cars.

NORTH JERSEY . . . Oversizing of pumps was the subject of a technical discussion prior to the regular meeting on October 5. J. Rogers of Bell & Gossett Company moderated. Main speaker of the evening was J. R. Lewis of Dunham-Bush, Inc., "Controlled Humidification."

UTAH . . . E. Case of Thiokol Chemical Corporation discussed use of commercial equipment in Thiokol operations in a talk presented at the September 22 meeting.

Guest speaker at the October 20 meeting was F. B. Kursh of Anemostat Corporation, who discussed "High Velocity Duct Systems."

BALTIMORE . . . Members of a special committee which investigated air conditioning in schools over the past year reported at the October meeting on what has been done to date throughout the country. Lester J. Welch, Director of School Facilities for Montgomery County, indicated that application of air conditioning to schools in this area has resulted in increased comfort and efficiency of students and teachers.

WESTERN MASSACHUSETTS . . . "Changes in the Construction Field" were detailed by October speaker W. R. T. Crolius of Dodge Reports.

RHODE ISLAND . . . Preparation and laying of the trans-Atlantic telephone cables between the United States and Europe was the subject of a film shown at the October 10 meeting.

SOUTH PIEDMONT . . . A simplified method of sound rating for air distribution equipment was detailed by September speaker D. Neviaser of Carnes Corporation.

W. W. Caines of Carrier Corporation, scheduled October speaker, was to discuss preventive and breakdown maintenance, with reference to various types of air conditioning and refrigeration equipment.

CENTRAL INDIANA . . . Moderating a discussion on application problems of refrigeration compressors, William Freije, Sr., headed a panel comprised of Edward Hunter of Duncan Supply Company, Walter Kinnard of Indiana Weathermaker and Charles Hottel of Langsenkamp. Although the open-type compressor can be applied over a wider range of suction pressures, it was stated, the hermetic unit has a number of advantages, such as lack of a rotary seal, often a cause of refrigerant leakage. However, hermetic units rarely are used under five ton. Although dehydration of every refrigeration system is considered essential, it was cited as being most important in hermetic units because of the enclosed motor being subject to corrosion resulting from excessive moisture.

When a choice of voltage is available, panel members decided, a higher voltage is better for larger machines because of economy in starter selection as well as current consumption. Service of 208 volt is satisfactory for units up to 20 ton, but 440 volt should be used whenever possible in large sizes.

Single-phase, 115-volt use is limited to quite small machines.

Questions from the membership reflected interest in increasing use of air condensers. With the emphasis on capacity control in cold weather, panellists concluded the best of several practical methods to be use of a modulating damper controlling air flow to eventually satisfy head pressure, possibly in conjunction with a two-speed fan motor. It was suggested further that a supplementary enclosure around the condenser could be helpful in preventing the ambient temperature dropping much below 40 F.

NEW ORLEANS . . . Subject of discussion at the September 20 meeting was "Bidding Rules Relative to the Mechanical Construction Industry." Panel members were: R. Putfork, C. Perrilliat, R. L. Autenreith and S. Prats. Following was a question and answer period on bidding and awarding rules and practices followed.

Following a tour of the LSU-NO Power Plant, as part of the October 17 meeting, Warren Moses of deLaureal and Moses discussed various aspects of the plant.

RICHMOND . . . Programs for future meetings were reported on at the September 7 meeting of the Executive Board by J. C. Turlington. Proposed for November 6 was a discussion of "Valve Sizing Applications" and "Economical Heating of Domestic Hot Water" is the topic for the January 8 meeting. Scheduled to replace the regular December meeting was a joint meeting on November 28 with the Illuminating Engineers Society. Following this, Milton Alley discussed committee memberships.

Considered at the Executive Board meeting on September 29 was the "Survey in Regard to National Dues and Allotments to Chapters," which is to be completed and returned to the Regional Director. Plans for the forthcoming Annual Joint Dinner Meeting of Engineers and Scientists was reported on by R. W. Sagnette.

Regional Director E. K. Wagner spoke at the regular Chapter meeting, held September 11, on results of the merger to date, problems which have arisen and overall prospects for the future of ASHRAE.

Use of natural gas engines for driving air conditioning equipment was discussed in three talks given at the October 2 meeting. J. Harmon of Carrier Corporation covered economics of the use of gas engines, Ralph Amos of Virginia Tractor Company detailed development and application of large gas engines and Thomas Lipscomb of Taylor, Lipscomb & Appel, Inc., discussed packaged units.

Region VI Meets in Detroit

Attending the Region VI Meeting, Life Member S. R. Lewis joined ASHRAE President John Everetts, Jr., First Vice President John H. Fox and Regional Director L. K. Warrick for a group portrait.

Michigan Chapter was host as delegates, alternates and guests gathered in Detroit October 16 for the Region VI Meeting.

Meetings ahead

January 25-26—Eighth Annual National Forum, Automotive Air Conditioning Association, Dallas, Texas.

January 29-February 1—American Society of Heating, Refrigerating and Air Conditioning Engineers, Semiannual Meeting, St. Louis, Mo.

February 9-11—Air Conditioning and Refrigeration Wholesalers, Annual Convention, Los Angeles, Calif.

February 12-15 – 12th Exposition of the Air Conditioning, Heating and Refrigeration Industry, Los Angeles, Calif.

March 3-8 – National Association of Frozen Food Packers, Chicago, Ill.

March 19-21 – National Electrical Manufacturers Association, Second National Electric Comfort Heating Symposium and Exposition, Chicago, Ill.

April 9-12 – National Oil Fuel Institute, 40th National Oil Heat Industry Convention and 24th Oil Heat and Air Conditioning Exposition, Chicago, Ill.

April 13-19—British Refrigeration Association, International Refrigeration Fair, London, England.

April 23-25—Gas Appliance Manufacturers Association, Annual Meeting, White Sulphur Springs, W. Va.

May 8-11 – Mechanical Contractors Association of America, 73rd Annual Meeting, Philadelphia, Pa.

May 22-23 – American Institute of Electrical Engineers, Domestic Appliance Conference, Columbus, Mo.

June 4-7—Institute of Boiler and Radiator Manufacturers, Annual Meeting, Absecon, N. J.

June 13-15—Heat Transfer and Fluid Mechanics Institute, 15th Annual Meeting, Seattle, Wash.

June 25-27—American Society of Heating, Refrigerating and Air Conditioning Engineers, 69th Annual Meeting, Miami Beach, Fla.

October 8-10—American Gas Association, Annual Convention, Atlantic City, N. J.

October 16-18 – American Standards Association, National Conference, New York, N. Y.

News of ASHRAE members

New jobs

J. R. Lewis has been appointed Manager of a new Dunham-Bush sales office established in Hillside, N. J., after having served with Carrier Corporation in various posts. Named Office Manager for the Buffalo sales office is **George T. Davis.** Previously Product Manager in West Hartford and Sales Engineer in Philadelphia, he had also been a Project Engineer for Tuttle & Bailey. **Donald F. Hardy,** an Auburn University graduate, is now Sales Engineer in the Birmingham, Ala., office. Newly appointed Sales Engineer in Greensboro, N. C., is **Winfred M. Kirkman.**

Richard J. Campbell recently joined Trane Company as a manufacturing engineer at the headquarters office in La Crosse, Wisc. He has had four years' experience as a manufacturing welding engineer and two years' experience as a welding research engineer with the U.S. Air Force, and is a graduate of Ohio State University.

Clifford J. Kimmel, appointed Special Project Engineer for C. V. Hill & Company, will be placed on special assignment with National Refrigeration Manufacturing, Inc., West Coast Div of Hill. A graduate engineer, he served previously with Dunham-Bush, Inc., most recently as Chief Product Engineer.

Leonard C. Bastian, former Chief Engineer—Air Conditioning for Philco Corporation, has been named Chief Engineer for both air conditioning and laundry. **Gabriel F. Moe,** formerly in charge of air conditioning product engineering, was named Assistant Chief Engineer—Air Conditioning. Succeeding Mr. Moe will be **William H. Mullin,** previously in charge of advanced design of air conditioners. **William J. Kramer,** an engineering specialist in air conditioning, was assigned to the advanced design post.

John W. Pulaski has been named Manager of the refrigerator and freezer cabinet section of the Westinghouse Electric Corporation Major Appliance Div engineering department. Joining Westinghouse after 14 years with General Electric Company, most recently as an engineering project engineer, he received his mechanical engineering degree from Syracue University in 1947.

Thomas J. Davidson, appointed Research and Development Engineer for Norcold, Inc., will be in charge of new product development. Prior to joining the company, he was a development engineer with Robertshaw-Fulton Controls Company for 15 years. Other affiliations were with the American Gas Association Laboratories and Rheem Manufacturing Company.

Charles W. Proctor, Jr., advances from his post as Commercial Sales Manager for the mid-Atlantic region for Minneapolis-Honeywell Regulator Company's Commercial Div to Director of Training. Succeeding him will be **Stanley Stake,** Philadelphia Branch Sales Manager for the past five years. **William J. Ortman,** appointed office building Market Manager, will continue also as Manager of Industrial and Marine Markets.

M. F. McCoy, former Manager of the refrigerator and freezer cabinet section of Westinghouse Electric Corporation Major Appliance Div engineering department, has been named Manager of the systems development and product evaluation section. He joined Westinghouse as a graduate student in the East Pittsburgh works in 1948, followed by advancement to Design Engineer in 1949 and Manager of the cabinet section in 1954.

Awards and recognitions

Charles W. Lockhart, Vice President in charge of sales for Buffalo Forge Company, has been elected President of the Air Moving and Conditioning Association, Inc. He has been with Buffalo Forge since receiving his B.S. in mechanical engineering from Cornell University in 1936. Other ASHRAE members elected to office of AMCA are: Vice President, **L. A. Macrow,** Chief Engineer, Carrier Air Conditioning Company; Vice President, **R. C. Niess,** Manager of Product Sales, York Div, Borg-Warner Corporation; and Secretary-Treasurer, **John F. Tobin,** Vice President of Marketing, American-Standard, Industrial Div.

James H. Downs, presently ASHRAE Region V Director, was presented on September 15 with a certificate by his employees to commemorate his 25 years as President of Refrigeration Supplies, Inc. Mrs. Downs looks on as Rod

O'Flaherty and William Polzer make the presentation. Beginning his career in 1923 with a Frigidaire distributor, Mr. Downs became Manager of Starr Company in 1936, purchasing Refrigeration Supplies in the fall of that year. He is a Charter Member of Refrigeration Service Engineers Society.

Necrology

John H. Shrock, deceased Life Member, had been Vice President of New York Blower Company.

James E. Hart, Consulting Engineer, died recently at the age of 36. He was a past-President of North Piedmont Chapter.

Gilray M. Anderson, Vice President of H. C. Gundlach Company, died September 16.

D. Kennard Smith was President of D. Kennard Smith, Inc., at the time of his recent death.

James F. Dailey, deceased, had been Chairman of the Board of Typhoon Heat Pump Company.

Gorton F. Price, President of Primore Sales, Inc., and General Manager of Universal Cooler Company, died recently at the age of 64.

Maxwell F. Rather, retired director of Johnson Service Company, died October 13 after a prolonged illness. An ASHRAE Life Member, he was 73. Active in Cleveland Chapter, he had served as Secretary and President. Nationally, he had been a member of the Nominating and GUIDE Committees.

BULLETINS

Recorders. Nineteen different types of single and multi-pen, ink and inkless, strip and round chart recorders in this line of direct and servo-operated, switchboard and portable recorders are listed in 12-page Bulletin GEA-6933A. For selection purposes, the recorders are described in accuracy classes of 1/4, 3/10, 1, 2 and 3%. Also covered are more than 40 different electrical and physical parameters that can be measured by these instruments in various utility and industrial applications. Information includes details of specifications, dimensions, applications, features, operating data and optional accessories.
General Electric Company, One River Rd., Schenectady 5, N. Y.

Processed Steels. Data sheets descriptive of aluminum-impregnated steels for high temperature, corrosive service are now available from this company. These steels are cited as having demonstrated a high degree of resistance to sulphur-bearing atmospheres at metal temperatures up to 1600 F and gas temperatures as high as 2200 F. A specific application of MLSCO-A/11 tubing and castings has been in superheater service for tubing and tubing supports.

Resistance to corrosion and oxidation is accomplished entirely by furnace impregnation at elevated temperatures, resulting in an iron-aluminum alloy layer ranging in depth as high as 0.035 in. There is therefore no coating which can peel, chip, crack or exhibit porosity.
M. L. Sheldon & Company, Inc., Corrosion Tubing Div, 350 Lexington Ave., New York 16, N. Y.

Portable Heaters. Comprising this new line are three heaters, the 107, 106 and 105. Two heat ranges are offered by the 107: 1200 and 1500 watt. Thermostat control permits dialed heat within each range. Model 105 is a compact heater serving in small enclosed areas. Flyer PH-6-61 is descriptive of the line.
Fasco Industries, Inc., Rochester 2, New York.

Brazing Stainless Steel. Four-page Brazing News #85 presents details of use of brazing alloys in joining stainless steel to itself and to dissimilar metals. Applications are illustrated.
Handy & Harman, 850 Third Ave., New York 22, N. Y.

Candidates for ASHRAE Membership

Following is a list of 180 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before December 31, 1961, of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

HAJEC, R. G., Devlpt. Engr., The Torrington Manufacturing Co., Torrington.
SNOBERGER, P. H., Sales Mgr., Robertshaw-Fulton Controls Co., Milford.

New Jersey

BONHAG, C. V., Pres., Northeastern Engineering Corp., Patterson.
BURAK, SOL, Sales Engr., Superior Steam Generators of N. J. Inc., E. Orange.
KRAMER, D. E.*, Service Mgr., Kramer Trenton Co., Trenton.

New York

ACCOMANDO, R. C., Junior Designer, Syska & Hennessy, New York.
FURMAN, R. W., Dir. Research & Development D/H Systems, Cargocaire Engineering Corp., New York.
HORNING, M. F., Engr., New York Telephone Co., Brooklyn.
JOHANSSON, H. R., Engr., J. W. Danforth Co., Buffalo.
KATKO, A. A., Sales Engr., Air Cond. Dist. Inc., Buffalo.
MCAULEY, J. J., Mgr. Acoustical Div, Insul-Coustic Corp., Maspeth.
MCCRAINE, THOMAS, Vice Pres.-Sales, Cargocaire Engineering Corp., New York.
MEYER, H. M., Sales Counselor, Knickerbocker Appliance Service Co., Albany.
OWENS, D. H., Mfg. Repr., A. L. Becker, Albany.
ROWLAND, W. B., Appl. Engr., Johns-Manville Sales Corp., Albany.
SMITH, C. D., Sales Engr., The Trane Co., Syracuse.
SULLWOLD, M. J.†, Project Mech. Engr., A. M. Kinney Associates, New York.

REGION II

Canada

ANDREW, JAMES, Project Engr., Galt Equipment Co. Ltd., Verdon, Quebec.
BEAUVAIS, JACQUES, Repr., Hamlet & Garmeau, Ltd., Montreal, Quebec.
CORSAIR, ROBERT*, Vice Pres., Durall Ltd., St. Boniface, Manitoba.
LEITCH, T. D., Chief Engr., A. S. Leitch Co., Ltd., Downsview, Ontario.
MACMARTIN, J. L., Com. Salesman, Honeywell Controls, Leaside, Ontario.
NUGENT, M. J.*, Mgr. A-C Dept., Canarctic Refr. Ltd., Ottawa, Ontario.
O'MORROW, L. A., Sales Engr., Alpha Manufacturing Co., Winnipeg, Manitoba.
PARKINSON, D. W., Sales Supvsr. Htg. Div, S. A. Armstrong Ltd., Toronto, Ontario.
PROCTOR, D. M., Sales Repr., Canadian Ingersoll-Rand Co. Ltd., Ottawa, Ontario.
RELVIK, MANIVALD, Design Engr., L. Villeneuve, Ltd., Toronto, Ontario.
ROSE, J. A., Estimator, B. A. Robinson Plbg. & Htg. Ltd., Scarborough, Ontario.
SALMON, G. T., Sales Repr., Jenkins Bros. Ltd., Toronto, Ontario.
SAVOIE, LEONARD, Plant Engr., Leo T. Julies, Quebec, Quebec.
STEVENS, J. F., Repr., Dominion Bridge Co., Montreal, Quebec.
SYLVESTER, R. K., Sales Mgr., Ont. A-C Div, R. H. Dyson Co., Ltd., Toronto, Ontario.

REGION III

Maryland

BUDD, R. E., Sales Engr., York Corporation, Baltimore.
SHREEVE, R. E., Engr., Egli & Gompf, Inc., Baltimore.
WEISMAN, W. J., Designer, Henry Adams, Inc., Baltimore.

Pennsylvania

DEANE, J. T., Regional Sales Mgr., Edwards Engineering Corp., Pittsburgh.
DESMOND, G. R., JR., Design Engr., Cronheim & Weger, Philadelphia.
GARABED, P. S., Partner, Remco Refrigeration Co., Philadelphia.
HARDIE, D. C., Mech. Engr., United Engineers & Constructors Inc., Philadelphia.
HARDIE, D. C., Mech. Engr., United Engineers & Constructors, Inc., Philadelphia.
MACALARNEY, G. C.*, Pres. & Engr., Soisson Inc., Cresson.

Virginia

ASBILL, C. M., III,† Field Engr., The Powers Regulator Co., Richmond.
HOUSE, G. A. JR., Contract Sales Engr., The Powers Regulator Co., Norfolk.
NEWSOM, A. H., JR., Asst. to the Pres., Portsmouth Gas Co., Portsmouth.
SLATER, W. H.†, Br. Mgr., The Powers Regulator Co., Richmond.

REGION IV

Florida

CERNI, E. M., Vice-Pres., Donaldson Electric Co., Inc., Jacksonville.

Georgia

AULD, J. A., Draftsman, Erickson's Inc., Savannah.
HUTCHINSON, D. M., Reg. Mgr., Westinghouse Electric Corp., Atlanta.
RAINWATER, J. H.*, Director of Engineering, The Warren Co., Inc., Atlanta.
ROGERS, C. M., JR., Chief Estimator, Chatham Pipe & Supply Co., Savannah.
SIKES, R. S., Sales Engr., Atlanta Gas Light Co., Atlanta.

North Carolina

BRISENDINE, J. D., Repr., Carrier Air Conditioning Co., Charlotte.
HUNTER, H. N., Owner, Southern Comfort, Charlotte.
KAUFMAN, E. E., Sales Engr., American-Standard, Ind. Div, Charlotte.
SPRINKLE, O. V., Engr., Chas. T. Main, Inc., Charlotte.

South Carolina

CASWELL, L. M., Estimator, W. O. Blackstone Co., Inc., Columbia.

REGION V

Indiana

EASTER, J. T., Resident Engr., The Powers Regulator Co., Fort Wayne.
KAMMAN, J. P.*, Methods & Layout Engr., Eli Lilly & Co., Indianapolis.
PIERCE, H. C.†, Major Devlpt. Engr., Arkle Air Conditioning Corp., Evansville.

Kentucky

WURTENBERG, O. A., Chief Engr., Continental Air Filters Inc., Louisville.

Ohio

BLOOMSBERG, T. N., Vice Pres., Bloomberg Plumbing Co., Inc., Youngstown.
DONNELLY, H. O., Pres., Harry O. Donnelly Sales Corp., Akron.
GALLAWAY, F. L., Engr., Antenucci, Inc., Warren.
HALIM, ABDELMESSIH, Design Engr., U. S. Air Co., Delaware.
HOPPER, P. H.*, Refr. Dept. Mgr., The W. H. Kliefaber Co., Dayton.
KRUSE, D. F., Mfg. Repr., Donald E. Delp Co., Columbus.
LERO, S. J., Partner, Lero & Markwald, Akron.
MAURER, R. A., JR., Residential Repr., Columbus & Southern Ohio Electric Co., Columbus.
MEYERS, B. G., A-C Specialist, Ohio Fuel Gas Co., Toledo.
NILSEN, LAURITS, Sales Engr., Tomlinson Steam Specialty Co., Cleveland.
PATTERSON, J. E., Appl. Engr., Harry O. Donnelly Co., Akron.
SHEARER, J. E., Chief Engr., Burt Mfg. Co., Akron.
SMART, E. J., Sales Repr., Columbus & Southern Ohio Electric Co., Columbus.
TOUSE, M. H., Sales Engr., Slawson Equipment Co., Canton.
WEEKS, J. R., Com. Sales Htg. & Cooling Specialist, Columbus Southern Ohio Electric Co., Columbus.
YEAGER, D. L., Associate Engr., Ohio Edison Co., Akron.

REGION VI

Illinois

ANDERSON, C. H., Associate, Earl Larson & Associates, Chicago.
BARR, F. J., Engr., Wilkins Pipe & Supply Co., Peoria.
HANSEN, R. C., Br. Mgr., The Powers Regulator Co., Peoria.
HARRIS, J. F., Vice-Pres., Staup & Harris, Chicago.
HOLBERT, K. T.*, Chief Engr., Murphy & Miller, Inc., Chicago.
LEAPLEY, A. O., Product Sales Mgr., Refrigerating Specialties, Chicago.
WARFEL, R. B., Project Engr., Skidmore, Owings & Merrill, Chicago.

Iowa

BUSWELL, G. M., Sales Prom., Interstate Air Conditioning Corp., Sioux City.
HOUVENAGLE, T. L., Designer Htg. Refr. A-C & Elec. Systems, Brooks-Borg, Des Moines.
MYERS, K. B., Assoc. Prof., Iowa State University, Ames.

Michigan

GRUBB, H. C., Product Mgr., American-Standard, Ind. Div, Dearborn.
LOEBL, R. A., Marketing Mgr., American-Standard, Ind. Div, Dearborn.
MIKULAS, WILLIAM,† Chief Engr., Kelvinator International Corp., Detroit.
MOORE, D. A., Sales Engr., R. L. Deppmann Co., Detroit.

Minnesota

MURPHY, C. P., Engrg. Supvsr., Minneapolis-Honeywell Regulator Co., Minneapolis.
PEARSON, R. C., Repr., McDowall Company, St. Cloud.
PERRY, L. R., Owner, Perry's Central Service, Minneapolis.

South Dakota

HANSON, A. E., Sales Engr., Shakstad Electric, Sioux Falls.
OLSEN, L. F., Owner, Mellgren Plumbing Shop, Inc., Rapid City.

Wisconsin

AHUJA, R. K., Trainee, Vilter Manufacturing Corp., Milwaukee.
DEBAUCHE, J. D.*, Htg. & Plbg. Designer, Mechanical Designer, Madison.

REGION VII

Alabama

BUCK, C. H., Owner, Bratton Sales Engineers, Birmingham.
ELLIOTT, H. M., Design Engr., Florence Heating & Air Conditioning Co., Birmingham.
KEITH, R. M., JR., Engr., Mobile Gas Service Corp., Mobile.
MCALEER, J. C., III, Sales Engr., Mobile Gas Service Corp., Mobile.

Kentucky

WEBB, W. M., Mgr. Engrg. Compressor, General Electric Co., Louisville.

Louisiana

LIGON, W. F., Sales Mgr., Universal Sheet Metal, Inc., Bossier City.
SMITH, LIONEL, Vice-Pres., Universal Sheet Metal, Inc., Bossier City.

Missouri

BARTELS, P. E.†, Mgr. Tech. Sales Service Dept., Alco Valve Co., St. Louis.
KEPLINGER, R. L., Sales Engr., Barber-Colman Co., Kansas City.
MORITZ, C. E., Vice-Pres., C. J. Moritz Co., St. Louis.
PALERMO, FRANK, Owner, Linden Heating & Air Conditioning, Kansas City.
PERUSICH, R. L., A-C Sales Engr., The Gas Service Co., Kansas City.
POTTS, J. D., Contract Sales Engr., The Powers Regulator Co., Kansas City.

Tennessee

COLMER, R. P., JR.†, Sales Engr., Yager & Nenon Inc., Memphis.
HUNT, C. W., Partner, Graves & Hunt, Memphis.

REGION VIII

Louisiana

FESS, M. D., Pres., Fess-Sieve & Co., Inc., Shreveport.
FRITH, D. C., Pres., Frith Sales, Shreveport.
JORDEN, J. C., JR.†, Assoc., Somdal-Smitherman - Sorensen - Associates, Shreveport.
ROBERTSON, B. B., Owner, Robertson Plumbing & Air Conditioning Co., Shreveport.

Oklahoma

NIKKEL, R. G.*, Chief Appl. Engr., Minneapolis - Honeywell Regulator Co., Oklahoma City.
WHITE, W. C., Repr., Gustin-Bacon Mfg. Co., Tulsa.

Texas

BAILEY, M. D., Repr., Minneapolis-Honeywell Regulator Co., Houston.
BALL, R. E.*, Sr. Appl. Engr., General Electric Co., Tyler.
BROWN, C. E., Sr. Engr., H. E. Bovay, Jr., Houston.
BROWN, M. W.*, Regional A-C Engr., American-Standard, Industrial Div, Dallas.
DIXSON, E. R., Pres., Dixson-Caperton, Inc., Tyler.
FLYNT, R. R., Owner, Mgr., Flynt Air Conditioning Co., Tyler.
GODWIN, J. C., Sales Engr., York Corporation, Houston.
LEE, J. E.*, Mechanical Consultants, Houston.
MCDANIEL, J. W., Sales Engr., The Trane Co., Houston.
MILSTEAD, C. F., Sales Engr., Toole & Co., Inc., Houston.
PRUITT, J. D., Sales Engr., The Trane Co., San Antonio.
RONAYNE, J. D., Replacement Specialist, General Electric Co., Tyler.
SHAFER, W. P., JR., Owner, Shafer Plumbing & Heating Co., San Antonio.
WALKER, W. E., JR., Engr., Jochen & Henderson, Houston.

REGION IX

Colorado

GRESSETT, T. G.†, Professional Engr., Littleton.

ASHRAE NATIONAL MEETINGS AHEAD

1962	1963	1964
Jan. 29-Feb. 1 Semiannual St. Louis, Mo.	Feb. 11-14 Semiannual New York, N. Y.	Jan. 27-29 Semiannual New Orleans, La.
June 25-27 69th Annual Miami Beach, Fla.	June 24-26 70th Annual Milwaukee, Wisc.	June 29-July 1 71st Annual Cleveland, Ohio

GUSTAFSON, R. A., Sales Engr., Industrial Service & Engineering Co., Denver.
SPROUL, J. C., Sales Engr., cba Company, Denver.

Montana

GAREN, D. R.*, Mech. Engr., Corp. of Eng. Ballistic Missile Constr. Office, Malmstrom A.F.B., Great Falls.

New Mexico

HELFRICH, WILLIAM, Architects & Engineers, Albuquerque.

South Dakota

REDLINGER, H. H., Pres., Redlinger Bros. Plumbing & Heating Co., Watertown.
STEWART, F. R., Pres. Owner, The Stewart Company, Rapid City.
TESSIER, E. A., Owner & Chief Engr., Tessier Sheet Metal Works, Mitchell.
WUERBEN, C. A., Technician, Johnson Service Co., Sioux Falls.
ZISKE, Plumbing Inspector, City of Sioux Falls, Sioux Falls.

Utah

LILLEY, J. L., Comm. Terr. Mgr., Lennox Industries Inc., Salt Lake City.
POUILLON, J. E., Asst. Sales Mgr., Comm., Lennox Industries, Inc., Salt Lake City.

REGION X

Arizona

CARLSON, A. C., Pres., Allen C. Carlson Co., Scottsdale.
DAVIS, H. C., Sales Engr., Norman S. Wright Co., Phoenix.

California

BACO, S. C., Mech. Engr., PW Engineering Div, UA Naval Air Facility, San Francisco.
BREESE, J. L.†, Editor & Publisher, Breese Publishing Co., San Francisco.
BREWSTER, H. W., Sales Project Engr., Fluor Products Co., Santa Rosa.
CHARLES, R. A., Partner, Charles & Braun, San Francisco.
GAFFNEY, L. H., Maint. Mgr., Scott Co., Oakland.
KOSSACOFF, E. L., Project Engr., Koebig & Koebig, Inc., Los Angeles.

WHO'S WHO IN ASHRAE

Insofar as possible these listings will each appear twice a year

ASHRAE OFFICERS, DIRECTORS COMMITTEES, STAFF

See page 80, November JOURNAL

REGION AND CHAPTER OFFICERS

See page 86, October JOURNAL

RESEARCH AND TECHNICAL COMMITTEES

See page 76, September JOURNAL

STANDARDS PROJECTS

See page 84, May JOURNAL

INTER SOCIETY COMMITTEES

See page 88, November JOURNAL

KULKA, SIEGMUND, Project Engr., Rodoni-Becker Co., San Francisco.
MAJER, G. L., Sales Engr., Master Fan Corp., Los Angeles.
MICKLITSCH, M. S., Sales Repr., The Trane Co., Los Angeles.
MILLER, R. R.*, Chief Engr., Thomas H. Parry, Pasadena.
MINER, R. E., Mech. Engr., District Public Works, San Diego.
SCHNEIDER, W. H., Mech. Engr., Robert B. Liles, Inc., San Francisco.
SHOWKEIR, J. M., Mech. Engr., Pacific Air Forces, HQ PACAF (PFSCE-ME), San Francisco.
SULLIVAN, P. D., Sales Repr., Burke & Co., Los Angeles.
TRAYLOR, J. E., Chief Mech. Engr., Bourne Associates International, San Francisco.
WRIGHT, G. R., Project Engr., Lawrence Radiation Laboratory, University of Calif., Berkeley.

Oregon

DRIVER, G. C., JR., Br. Sales Mgr., Carrier Corp., Portland.

Washington

CROSS, B. W., Sales Engr., Dunham-Bush, Inc., Seattle.
MCAFEE, R. L.†, Br. Mgr., Barber-Colman Co., Seattle.
PROCTOR, J. F., Secy.-Treas., E. Sam Dick Co. Inc., Seattle.
SWANK, D. H., Design Specialist, Boeing Co., Renton.
THORGAARD, CLIFF, Pres., Thorgaard Plumbing & Heating Co., Inc., Seattle.

Hawaii

SODERHOLM, N. L. JR., Principal & Chief Engr., Montgomery & Associates, Honolulu.

FOREIGN

England

KIDD, A. A., Chief Design Engineer, J. G. L. Paulson, L/Friba, Yorkshire.

STUDENTS

ANEKWE, G. C., Purdue University, Lafayette, Ind.
CORDER, D. L., Purdue University, Lafayette, Ind.
DOHERTY, W. G., Purdue University, Lafayette, Ind.
GARABED, E. P., Lehigh University, Bethlehem, Pa.
GRAFT, J. T., Purdue University, Lafayette, Ind.
HERDER, J. K., Purdue University, Lafayette, Ind.
JOHNSON, R. J., Purdue University, Lafayette, Ind.
KEEGSTRA, G. V., Purdue University, Lafayette, Ind.
MAY, D. L., Purdue University, Lafayette, Ind.
ROBINSON, P. E., Purdue University, Lafayette, Ind.
STRIKIS, JANIS, Purdue University, Lafayette, Ind.

NO MORE DEFICITS

(*Continued from page* 52)

Committee. What are the additional benefits to our members to justify this further increase of income by elimination of chapter allocations? The major benefit will be to permit the chapters to have complete autonomy with relation to chapter members. At present, under the allocation system, members of the Society are assigned a chapter by the headquarters office. As approximately 65% of all chapters assess chapter dues, there is an increasing dissatisfaction among these chapters, especially when an assigned member refuses to carry his share. A second benefit would be a reduced cost of operations in the headquarters office which is now necessary to maintain and constantly correct membership records and reassignment of members to chapters in order properly to provide this allocation. It is a safe assumption that, due to inflation, in a few years the same services will cost more. And in the meantime we should add to our meager reserves.

On June 30 of this year all Society funds amounted to $773,000. Almost 60% of these funds is earmarked for research, and the magnitude of the Society reserve fund is but $176,000. Assuming all our other resources had been exhausted by an emergency, this sum would merely cover our present operation, exclusive of research grants, for a period of 10 weeks. That is woefully inadequate and underscores the real need for accumulating additional reserve funds. We can do this by increasing our membership, by the sale of more advertising in our publications and by the proposed elimination of chapter allowances.

Air Flow Problems are NWAHACA Topic

Convening in Chicago, the National Warm Air Heating and Air Conditioning Association held its 48th Annual Convention November 8, 9 and 10. Delivering the opening address, NWAHACA President H. C. Gurney presided over the preliminary session Wednesday afternoon, topic for which was Association Affairs and Management. During this session, manufacturer-members were given an opportunity to discuss industry problems and needs, providing the Board of Trustees with members' viewpoints on Association activities.

Reviewing accomplishments of the Association during 1961, H. P. Mueller, Jr. (Associate ASHRAE), Second Vice President of NWAHACA and President of Mueller Climatrol Div, Worthington Corporation, spoke on "What We Have Done in '61." NWAHACA First Vice President Don Winegardner, Vice President of the Heating Div, Majestic Company, followed up with "What We Can Do in '62," outlining proposed programs and how they will further the Association's goals.

Two simultaneous sessions open to all industry members were held on Thursday, covering "Technical and Application Engineering" and "Marketing." Otto J. Ress (Member ASHRAE), Manager of Engineering, Mueller Climatrol Div, Worthington Corporation, and Chairman of the NWAHACA Research Advisory Council, presided over the morning Technical and Application Engineering Session. Papers were presented by Murray Patterson of the University of Illinois, "Does Modulation Improve Comfort Performance?"; Kendall H. Flint, Director of Technical Service, Gas Vent Institute, "Vents . . . Their Selection and Design"; James H. Healy (Associate ASHRAE) of the University of Illinois, "When Winter Winds Blow," an evaluation of results of studies conducted to determine how and why infiltration occurs; and T. R. Casberg, Staff Specialist for Mechanical Equipment and Systems, U. S. Department of Defense, "Development of New Military Housing Specifications."

Presiding in the afternoon was Lorin G. Miller (Member ASHRAE), Consulting Engineer and Chairman of the NWAHACA Application Engineering Council, who additionally presented a paper reviewing results and problems encountered with improperly sized equipment. Other speakers were Dr. Jay Spiegleman, Albert Einstein Medical Center, "Air Processing in Respiratory Allergies"; Boris Hryekewicz of the University of Illinois, "How Accurate Are Our Heat Loss Calculations?", in which he discussed findings of a study of measured energy consumption compared with calculated heat losses for the University's Research Residence No. 4; and Herbert T. Gilkey (Member ASHRAE), NWAHACA Director of Technical Services, "Introducing the New Manual."

During the morning session of the Marketing Conference, at which Don Winegardner presided, Kentner L. Wilson (Associate ASHRAE), Vice President of Minneapolis-Honeywell Regulator Company, presented a filmed report of a seven-city survey of consumer attitudes towards the heating and air conditioning industry. Among topics covered were products and methods of selling, installing and servicing.

Guest speaker at the luncheon session was Richard J. Cannon, Assistant Commissioner, Federal Housing Administration.

Afternoon session of the conference, presided over by H. P. Mueller, Jr., featured talks on the relationship of the "Real Buyer" to: the manufacturer – A. W. Beck, Vice President, Western Operations, Robertshaw-Fulton Controls Company; the wholesaler –J. Orville Garrett, Vice President of Loman Garrett Supply Company; the utilities–Chester S. Stackpole (Associate ASHRAE), Managing Director of the American Gas Association; and the dealer–Randall A. Nelson, NWAHACA Director of Public Relations. Concluding the program was a summation and evaluation of the major facts presented throughout the conference, discussed by J. P. Field, Vice President, Williamson Company.

Theme of the Region Three Dealer Conference on Friday was "Organizing, Operating and Selling for Profit." Carl W. Millsom (Associate ASHRAE), Vice President–Sales, Perfection Div, Hupp Corporation, and Chairman of the NWAHACA Trade Relations Committee, presided. NWAHACA President H. C. Gurney delivered the welcoming address. Size and scope of the selling job in local markets were indicated in a filmed report, following which NWAHACA Assistant Director of Public Relations James D. Nolan covered fundamentals of selling quality systems.

Highlighting the morning session was establishment of the first of eleven Regional Dealer Councils, with election and installation of Region Three members. The chairman of each Regional Council automatically will become a member of the National Dealer's Advisory Council. Through these two bodies, dealer-contractor members of NWAHACA will have a direct means of expressing their specific needs to the Association.

J. P. Field presided over the afternoon session, which featured a step-by-step report of operation of an Indoor Comfort Bureau and promotion of Silver Shield systems, presented by Glen W. Rynbrand (Member ASHRAE) of the Glen W. Rynbrand Company, Executive Secretary of the Kalamazoo Indoor Comfort Bureau. Randall A. Nelson, James D. Nolan and Robert A. LaChance were concluding speakers of the Convention, covering in detail what the dealer-contractor business is and does, duties of management, resources of management, income transactions, credits and collections, expense transactions and estimating and pricing for profits.

An advance view of technical sessions, symposiums and other program events planned for the Semiannual Meeting, St. Louis, January 29-February 1, will appear in your January JOURNAL.

ASHRAE JOURNAL

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More cooling in the same equipment

Compared with "Freon-22", "Freon-502" increases cooling capacity 10% at –10°F and 12% at –20°F. At still lower evaporating temperatures even greater capacity can be expected.

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The compressor discharge temperature with "Freon-502" is some 40°F lower than for "Freon-22". High discharge temperatures have long been recognized as contributing to instability of the oil-refrigerant system and resulting motor burnouts.

Below is a comparison of discharge temperatures at different evaporating temperatures:

DISCHARGE TEMPERATURES

Evaporator Temperature	–20°F	–10°F	+40°F
"Freon-502"	238	226	162
"Freon-22"	282	272	191

"Freon-502" is just another example of the forward thinking you expect . . . and get . . . from Du Pont, the pioneer and leading manufacturer of refrigerants. We'll be happy to work closely with you in engineering your equipment for "Freon-502". Du Pont Company, "Freon" Products Division, N-2420, Wilmington 98, Del.

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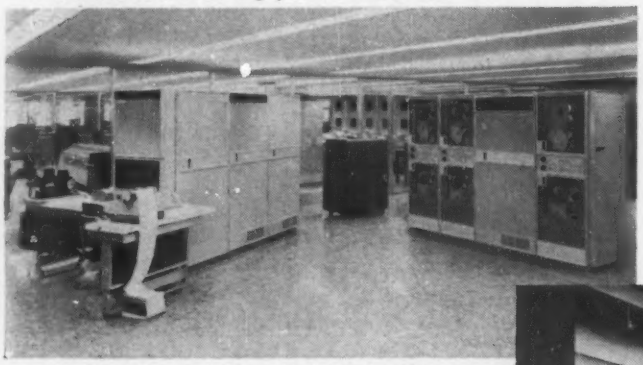
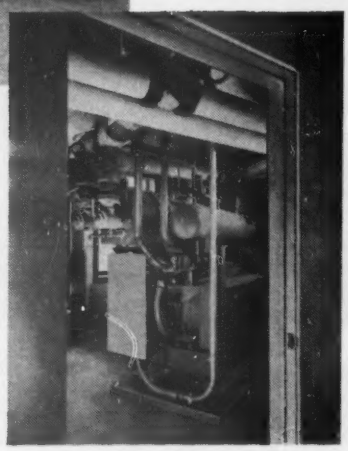
"Hot" Brain requires Cool Treatment

To process 2,400 checks a minute and provide fast service for its 75,000 checking accounts, the First Wisconsin National Bank of Milwaukee is spending more than $1,500,000 for an automated check handling system second to none.



▲ A view of the First Wisconsin National Bank's new automated check processing system currently geared to handle about 500,000 checks a day. Shown is the computer room. Sorter room is to the left. A third area, the printer room, completes the installation.



This Vilter Uni-Chiller maintains the required temperature with a capacity of 61.8 tons with 55° entering water and 45° leaving water. ▶

● Currently programmed to process the avalanche of 500,000 checks a day . . . 2,400 a minute . . . is the new, fully expandable automated "brain" check handling system built especially for the First Wisconsin National Bank of Milwaukee.

The "hardware" used in this amazing system—computers, high speed magnetic tape machines and printers, multiplex control, document sorters and check encoders—generate substantial amounts of heat which must be rapidly dissipated to protect both the components themselves and the accuracy of the processing.

To help maintain the critical temperature and humidity levels within the glass-enclosed 8,000 square foot area, a Vilter UF-6, flooded type, R-22 Uni-Chiller was installed. Cooling coils are mounted in a plenum chamber with air distributed into the conditioned area through a slotted acoustical tile ceiling. This method of air distribution produces an imperceptible movement of air for maximum comfort of personnel in the room.

You benefit with Vilter refrigeration and air conditioning equipment in two important ways—(1) careful design and precision fabrication result in long life . . . reliable, efficient performance . . . minimum maintenance; (2) a complete and integrated line of equipment, and the experience to counsel you on the best type of installation. Why not let the nearest Vilter representative help you.

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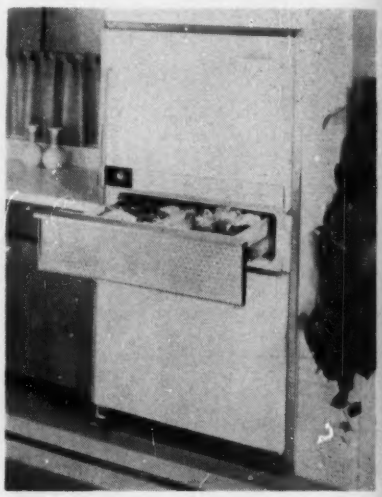
NEW PRODUCTS

MAJOR APPLIANCE LINES

Half the twelve models in the refrigerator line are frost-free and seven use a cold injector refrigerating system. Two models feature a center drawer (shown) for storing fresh meats and vegetables. Added to the line is



the Spacemaster, with more than 90% of its 12.6 cu ft for fresh food storage.

Capacities of the freezer line range from 333 to 740 lb. Each model features low running time, quiet operation, below zero freezing and low operating costs. Air sweep shelves in upright models permit greater air circulation.

Expanded, the dehumidifier line is highlighted by the Custom Supreme combination dehumidifier and space heater, which has a 1300-watt heater. Two heavy-duty models can remove up to six and eight gal of moisture a day.

Two models in the room air conditioner line adjust automatically to meet sudden or gradual temperature changes. Mobilaire units have a cart for moving them from room to room. **Westinghouse Electric Corporation, Columbus, Ohio.**

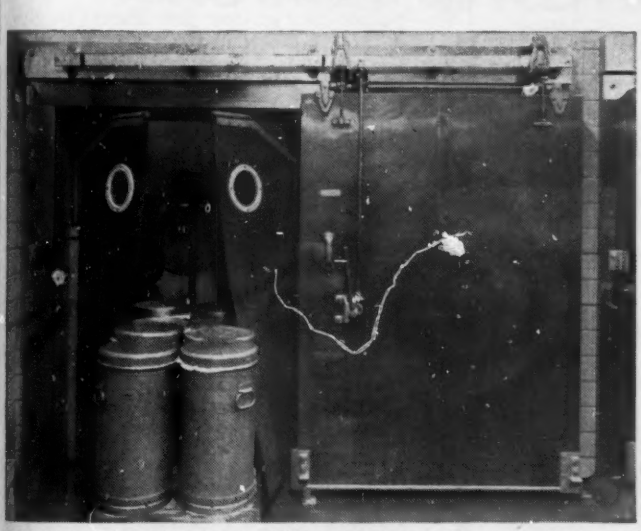
WALL FURNACES

Heavy-duty, dual centrifugal blowers on Forsaire counterflow, gas-fired wall furnaces deliver up to 430 cfm of warm air. Motor and blower wheels are suspended on vibration isolators. Other features include 24-volt automatic gas valve, twin cast iron burners with raised ports and die-formed, ceramic coated heat exchanger.

Units are offered in sizes from 35,000 to 65,000 Btu/hr input. All models are approved for installation recessed between studs or as free-standing wall units. Multiple heat

FAST OPENING AND CLOSING. Jamison Electroglide® Power Doors speed traffic to loading dock—single leaf cooler door, left, and bi-parting freezer door, right.

New $3,000,000 automated plant features Jamison Horizontal Sliding Doors



SAVES REFRIGERATION. Jamison Manual Horizontal Sliding Door in combination with self-closing Flexidor®.

RAPID, CONVENIENT LOADING. Jamison Manual Horizontal Sliding doors facilitate loading of dairy products.

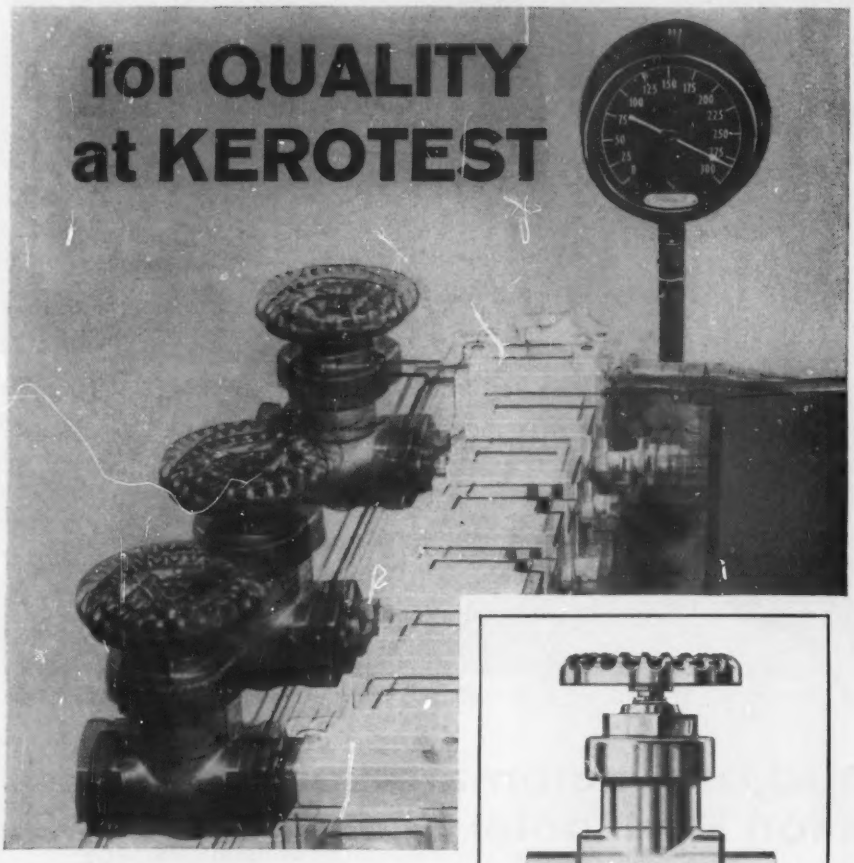
• In a dramatic advance toward automated processing of fluid milk and ice cream products, the Borden Company recently completed a new $3,000,000 plant at Milwaukee, Wisconsin. An important feature of this modern plant is a number of Jamison Horizontal Sliding Cold Storage Doors.

More and more, in the design of new, modern facilities like this one, Jamison Doors are specified to save space, minimize refrigeration loss, speed traffic and facilitate handling of products and materials. Get the story of the cost-saving benefits of Jamison Doors from your architect, or write to Jamison Cold Storage Door Co., Hagerstown, Md.

JAMISON

COLD STORAGE DOORS

SHAKEDOWN



Vibration-proof Construction

Only precision, quality-controlled construction can take this kind of concentrated abuse. These valves, under a pressurized refrigerant, are vibrated at 3600 cycles per minute for prolonged periods. At intervals throughout the accelerated test, Kerotest engineers evaluate performance and condition of seat seals, diaphragm seals, solder seals and handwheels.

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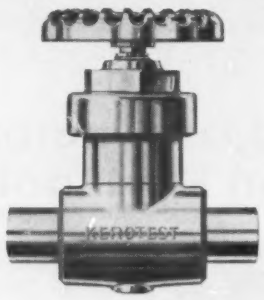

KEROTEST 520 DIAPHRAGM PACKLESS GLOBE VALVE

The precision-built brass valve for all types of refrigeration and air conditioning systems. Ideal for liquefied petroleum gas bulk plants, pipe line loading racks, laboratory installations, nitrogen, oxygen (degreased) and compressed air applications.

Specify the Kerotest 520 globe valve for operating pressures up to 400 psi and temperatures up to 200° F. Use it in vacuum service applications to 1 micron.

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- Replaceable spindle seats for continuous positive seating
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- Generous handwheel

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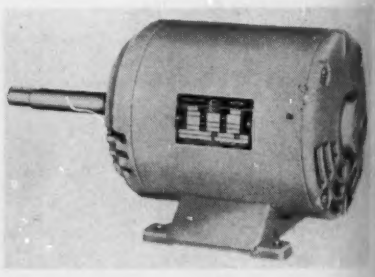
outlets provide one, two or three-room heat delivery.
Williams Furnace Company, Buena Park, Calif.

AIR CONDITIONER

Combining the flexibility of the split-system air conditioner with factory-sealed advantages and installation ease of integral units, the Quik-Attach air conditioner can be installed outdoors in any convenient location. The cooling coil, after being mounted in place, is connected to the outdoor unit with pre-charged tubing, available in several lengths. Featured is the Spine-Fin condensing coil, comprised of many tiny aluminum spines, cited as being up to 50% more efficient than plate-fin coils. Standard equipment includes the air-cooled remote condensing unit, pre-charged connecting tubing with Quik-Attach Valves, filter-drier, low-pressure cut-out, two-speed fan motor with permanently lubricated sealed bearings and cooling coil for indoor installation with either upflow or downflow furnaces.
General Electric Company, Central Air Conditioning Dept., Troup Hgwy., Tyler, Texas.

PUMP MOTORS

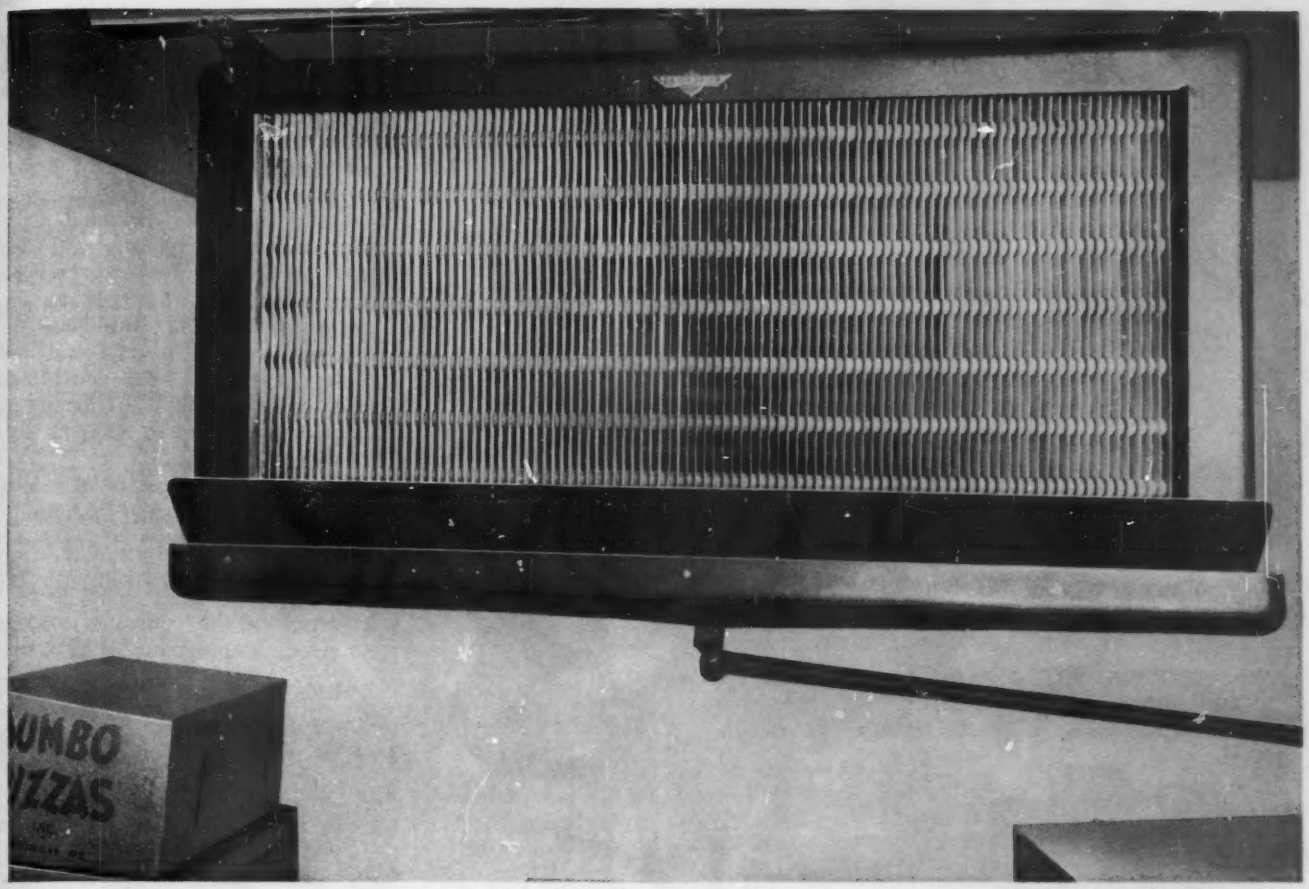
Two groups of motors, developed for pump duty service, are offered by this manufacturer: jet pump, 56-frame motors from ¾ to 2 hp and close-coupled,



182 to 254U frames in ratings to ten hp. Both designs are available in single-phase capacitor and three-phase squirrel cage types for high starting torque, low starting current, constant speed operation.
Kingston-Conley, Inc., Sub of Howell Electric Motors Company, 16316 W. Seven Mile Rd., Detroit 35, Mich.

DUCT FURNACES

Gas duct furnaces for installation downstream of refrigeration coils have low air pressure drop and are designed for use with air conditioning systems where by-pass ducts would be too bulky. Moisture which forms inside the heat exchanger tubes in summer is diverted by a stainless



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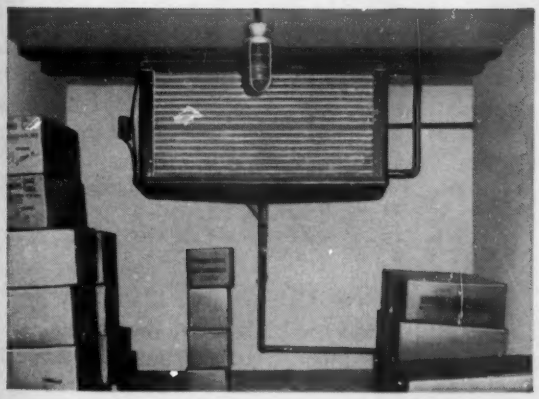

Low initial cost, simplicity of installation, good service—these are the points that earned LARKIN Low Temperature Humi-Temps a reputation throughout the free world.

HOT-GAS DEFROST

Standard LT units, mullion units and wall units are available with LARKIN's patented Frost-O-Trol® Hot-Gas Defroster. It is simple, effective, fool-proof, completely automatic. A factory-installed metering orifice prevents slugging of refrigerant into compressor—making extra tanks, reversing valves or re-evaporators unnecessary.

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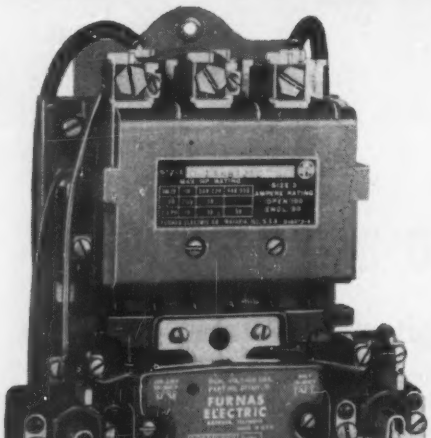

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steel condensate shield away from the pilot, thermocouple and ignition slots and into an aluminized steel bottom pan with threaded drain connection.

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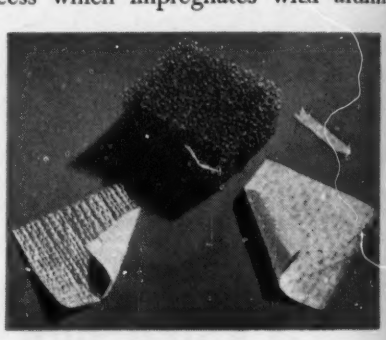
Hastings Air Control, Inc., 3215 Leavenworth St., Omaha 5, Nebr.

ALUMINIZING PROCESS

Developed by this company is a process which impregnates with alumi-



num textiles, leather, paper or plastics, adding heat resistance to a cited 2500 F, depending on the base material, while retaining texture and pliability of the base material. Alumination (as the process is designated) may be applied to one or both sides of a fabric, and depth of coating is controlled closely.

Thermo-Chem Corporation, Noeland Ave., Penndel, Pa.

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For varied industrial air handling applications, SQ airfoil units are offered in sizes from 12¼ to 18¼ in. in single and double inlets. Static pressures are from ¼ to 9 in., and air volume ranges to 15,000 cfm with temperatures to 200 F.

Chicago Blower Corporation, Franklin Park, Ill.

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phragm for high resistance to stem wear and a lower diaphragm of stainless steel to protect the upper assembly against corrosion. Valves are available in most tube sizes, with angle or two-way flare pipe or solder connections.
Imperial-Eastman Corporation, 6300 W. Howard St., Chicago 48, Ill.

BASEBOARD ELEMENT

Incorporating four tracks of heat-resistant polypropylene, the Panel-Track element offers reduction in element damage due to the fact that the track clipped on the corners of each fin gives the element a rigid and rugged boxlike structure. These tracks serve also as sound dampers.
Spencer Chemical Company, Dwight Building, Kansas City 5, Mo.

DEHUMIDIFIERS

Three of these four new models feature a built-in humidistat, which automatically turns the unit on and off to maintain the desired humidity level. The completely automatic Custom Dehumatic DHF-5 model (shown) in-

cludes the Safety Flowguard, which turns the unit off when the condensate pan is full. Units are equipped with concealed, ten-qt, rust-proof polyethylene condensate pans. Compressors, with capacities of 1/5 and 1/6 hp, are cited as removing up to 38 and 48 pints of moisture, respectively, per 24 hr. Equipped with casters, all models may be wheeled from room to room or may be connected to permanent drains.
Fedders Corporation, 58-01 Grand Ave., Maspeth 78, N. Y.

CENTRIFUGAL FANS

Direction flow wheels are among improvements incorporated in the Design 6 series of tubular centrifugal fans. Available for volumes up to 300,000 cfm and static pressures to 15 in., these in-line fans are built in Class I, II, III, IV and V constructions for high pressure and conventional air conditioning, ventilation and indus-

trial process applications. Units have aerodynamically designed centrifugal wheels with die-formed blades and shroud. The wheel imparts centrifugal motion to the air and then diverts it to axial flow within the impeller, minimizing turbulence.
Dryer Dynamics Corporation, Page & Schuyler Aves., Lyndhurst, N. J.

FORCED AIR HEATERS

Gas-fired, the Radiantflow heater has a maximum output of 40,000 Btu/hr, for applications where characteristics of infra-red heating offer the greatest advantages, such as work areas of large buildings with low occupancy, areas where air movement is not desirable, heating machines, process drying and outdoor work areas.

In the heater, a rectangular, unvented, open-flame unit, combustion occurs on the enclosed surface of a high temperature alloy screen. When the metal reaches maximum temperature, infrared rays radiate from its exposed surface. An aluminum reflector concentrates the rays on the area being heated.
Dravo Corporation, Neville Island, Pittsburgh 25, Pa.

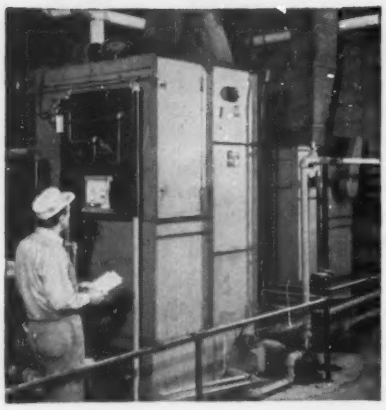
NINE AIR CONDITIONERS

Designated the air jet vane line because of the air delivery system based on more powerful blowers and an air

jet vane front grille. Flexibility of the grille permits either wide disper-

DRY and CLEAN AIR at the RIGHT TEMPERATURE

- to control your product quality
- to protect a critical operation
- to protect apparatus from moisture damage
- to DRY your material or product
- to control packing or storage conditions
- to assure precision in testing or research
- to increase air conditioning capacity



Air Condition by the NIAGARA Method Using HYGROL® Liquid Absorbent

This compact method, giving high capacity in small space, removes moisture from air by contact with a liquid in a small spray chamber. The liquid spray contact temperature and the absorbent concentration, factors that are easily and positively controlled, determine exactly the amount of moisture remaining in the air.

Most effective because... it removes moisture as a separate function from cooling or heating and so gives a precise result, and always. Niagara machines using liquid contact means of drying air have given over 20 years of service. The apparatus is simple, parts are accessible, controls are trustworthy.

Most reliable because... the absorbent is continuously reconcentrated automatically. No moisture-sensitive instruments are required to control your conditions... no solids, salts or solutions of solids are used and there are no corrosive or reactive substances.

Most flexible because... you can obtain any condition at will and hold it as long as you wish in either continuous production, testing or storage.

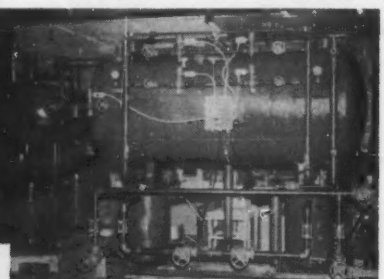
Write for Bulletins 112 and 131 and complete information on your air conditioning problem.

NIAGARA BLOWER COMPANY

Dept. RE-12, 405 Lexington Ave., New York 17, N.Y.

Niagara District Engineers in Principal Cities of U. S. and Canada

THE ULTIMATE - IN THE COMPRESSION REFRIGERATION CYCLE*



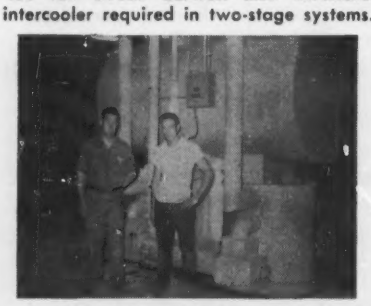
THIS IS ANOTHER CYCLE CENTER, factory assembled and on its way to a 400 ton milk and ice cream plant.

This 400 ton CYCLE CENTER also eliminates the intercooler required in two-stage systems.

WHAT WILL IT DO?*

It will provide liquid overfeed to the evaporators, catch the excess liquid and recirculate it to the evaporators, with these results:

- FULL COMPRESSOR PROTECTION AGAINST SLUGS
- PEAK COIL AND COMPRESSOR EFFICIENCIES
- SUB COOLED LIQUID FEED AT CONSTANT PRESSURE THE YEAR AROUND
- PRACTICALLY UNLIMITED RATE OF LIQUID FEED AT ABSOLUTELY NO POWER COST
- NO MECHANICAL PUMPS
- NO FLASH GAS IN LIQUID LINES
- SAFE, AUTOMATIC PLANT OPERATION
- OIL SEPARATION, ANY REFRIGERANT
- HIGHER SUCTION PRESSURES
- LARGE POWER SAVINGS



A similar 400 ton CYCLE CENTER after installation and insulation.

- LARGE SAVINGS IN FIRST COST ON NEW PLANTS. FOR EXAMPLE, THE RECEIVER IS NOT REQUIRED AND SURGE DRUMS ARE ELIMINATED.
- AUTOMATIC HOT GAS DEFROSTING AT MINIMUM COST.

ASK FOR BULLETIN CC-2

*NOT JUST A LIQUID RETURN UNIT.

Available for any refrigerant, in capacity from 10 to 1,000 tons and more. Factory assembly is optional.

J. E. Watkins Co.

1311 SOUTH FOURTH AVENUE
MAYWOOD, ILLINOIS

sion or concentration of cooled air. Each unit has a two-speed fan. Accordion-like mobile frame comes assembled and can be fitted into windows ranging from 28 to 48 in. wide. Mobile cart assembly permits easy movement of the conditioner from room to room.

Included in the line are two heavy-duty units that automatically adjust their operation to meet sudden or gradual changes in temperature. Units in the MRB series range from 6100 to 9500 Btu/hr, heavy-duty units from 9500 to 16,500 Btu/hr. **Westinghouse Electric Corporation, Columbus, Ohio.**

SIX-IN. THERMOMETERS

Adjustable to any desired viewing angle, six-in. scale Multiform industrial thermometers are cited as simplifying installation. Positive locking at the selected position is assured by internal tapered lock construction and a set screw adjustment. Thermometers are available with separable wells for liquid service or with bare bulbs and perforated stems for greater sensitivity and speed of response in indication of air temperatures. Ranges are from −40 to 500 F. All models have a red reading mercury column and white face with black markings. **Precision Thermometer and Instrument Company, 1434 Brandywine St., Philadelphia 30, Pa.**

COOLING SYSTEM

(*Continued from page* 63)

ity (grains moisture/lb dry air)

G_w = absolute humidity at winter design space relative humidity for the Radiant Air System (grains moisture/lb dry air)

L_{ms} = maximum allowable occupancy design summer latent load (Btu/hr/ft²)

$(\Delta S)\ T_r$ = Increase in entropy for building cooling system removing thermal load at T_i and rejecting it to heat transfer media at T_r

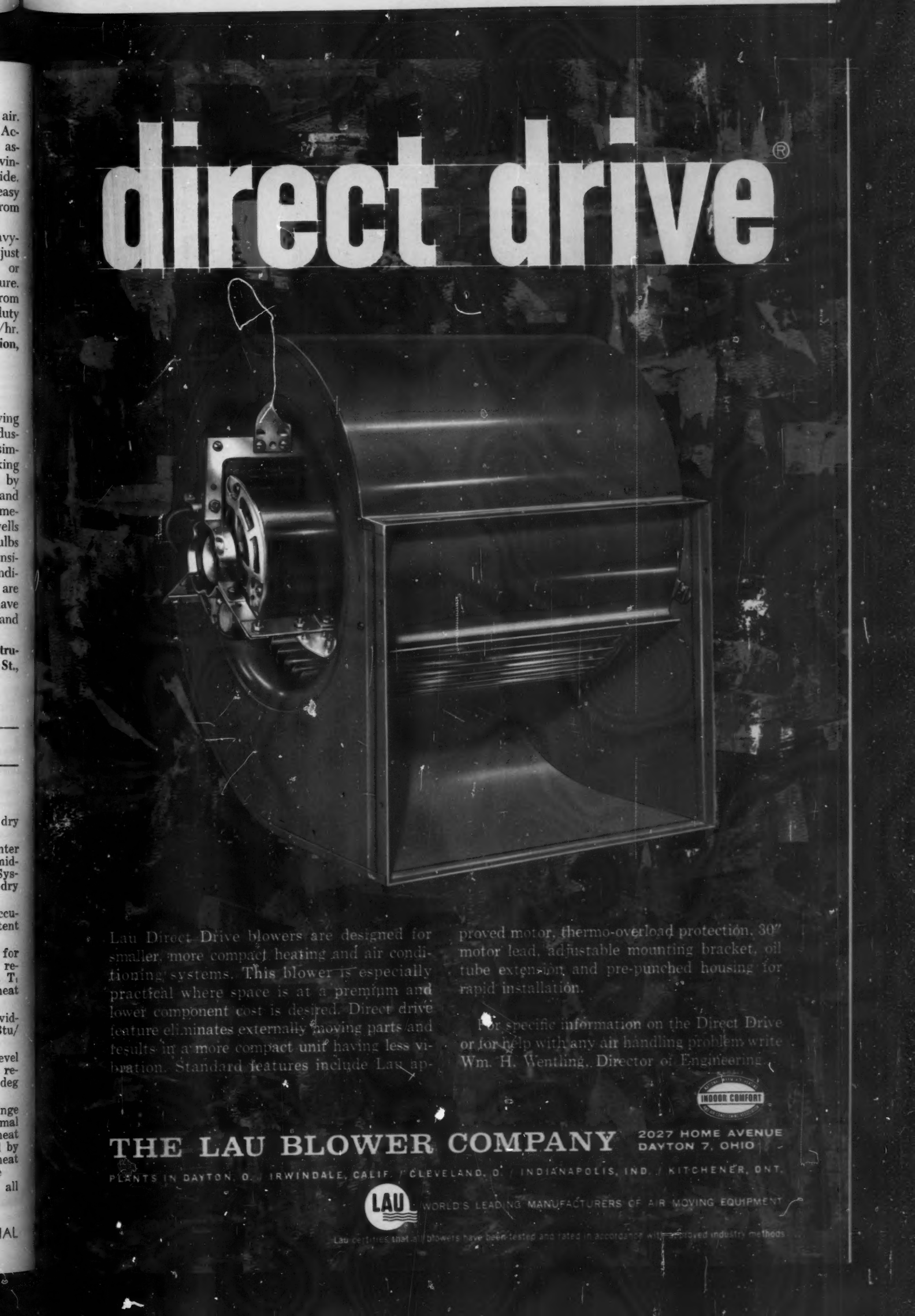
Q_i = Thermal load from individual heat sources (i)—Btu/hr/ft²/F

T_i = Temperature exchange level at which thermal load is removed by system — deg Rakine

T_r = The temperature exchange level at which all thermal load from individual heat sources (i) are rejected by system to external heat transfer media—Rakine

Q_r = The algebraic sum of all loads designated by Q_i

direct drive®



Lau Direct Drive blowers are designed for smaller, more compact heating and air conditioning systems. This blower is especially practical where space is at a premium and lower component cost is desired. Direct drive feature eliminates externally moving parts and results in a more compact unit having less vibration. Standard features include Lau approved motor, thermo-overload protection, 30" motor lead, adjustable mounting bracket, oil tube extension and pre-punched housing for rapid installation.

For specific information on the Direct Drive or for help with any air handling problem, write Wm. H. Wentling, Director of Engineering.

INDOOR COMFORT

THE LAU BLOWER COMPANY

2027 HOME AVENUE
DAYTON 7, OHIO

PLANTS IN DAYTON, O. / IRWINDALE, CALIF. / CLEVELAND, O. / INDIANAPOLIS, IND. / KITCHENER, ONT.

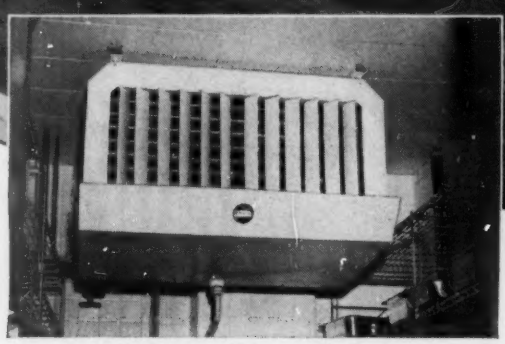
LAU — WORLD'S LEADING MANUFACTURERS OF AIR MOVING EQUIPMENT

Lau certifies that all blowers have been tested and rated in accordance with approved industry methods

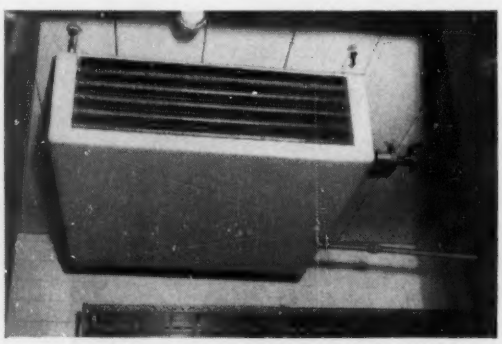
• *REFRIGERATION CONTRACTORS: Refrigeration Sales & Engineering* • *WHOLESALER: Jones-Newby Supply Co.*

CERTIFIED CAPACITIES, RELIABILITY, SERVICE

decided Western Electric's choice of **KRACK REFRIGERATION UNITS** *for new Oklahoma City Plant.*



8 walk-in storage rooms, including 2 freezers, all equipped with Krack units, maintain just the right climate for the class of food stored.



Krack unit in salad refrigerator has a special hot-tin-dipped coil for protection against corrosive action of acids in salads.

Krack equipment relied upon exclusively to keep food garden-fresh in new plant cafeteria.

Each class of food stored—dairy products, fruits and vegetables, meats, salads and bakery goods—needs exact temperature and humidity control to keep food at its peak of flavor and freshness. Krack equipment fully meets these exacting requirements—as expected.

FREE BULLETIN 620 gives data on reliable low or high temperature refrigeration equipment at production unit cost.

Mail coupon or phone today!

KRACK ENGINEERED

REFRIGERATION APPLIANCES, INC.

Manufacturers of Freon, or Ammonia, Recirculated, Flooded or Direct Expansion Heat Transfer Equipment.

Refrigeration Appliances, Inc.
917 Lake Street, Chicago 7, Ill • MOnroe 6-1141

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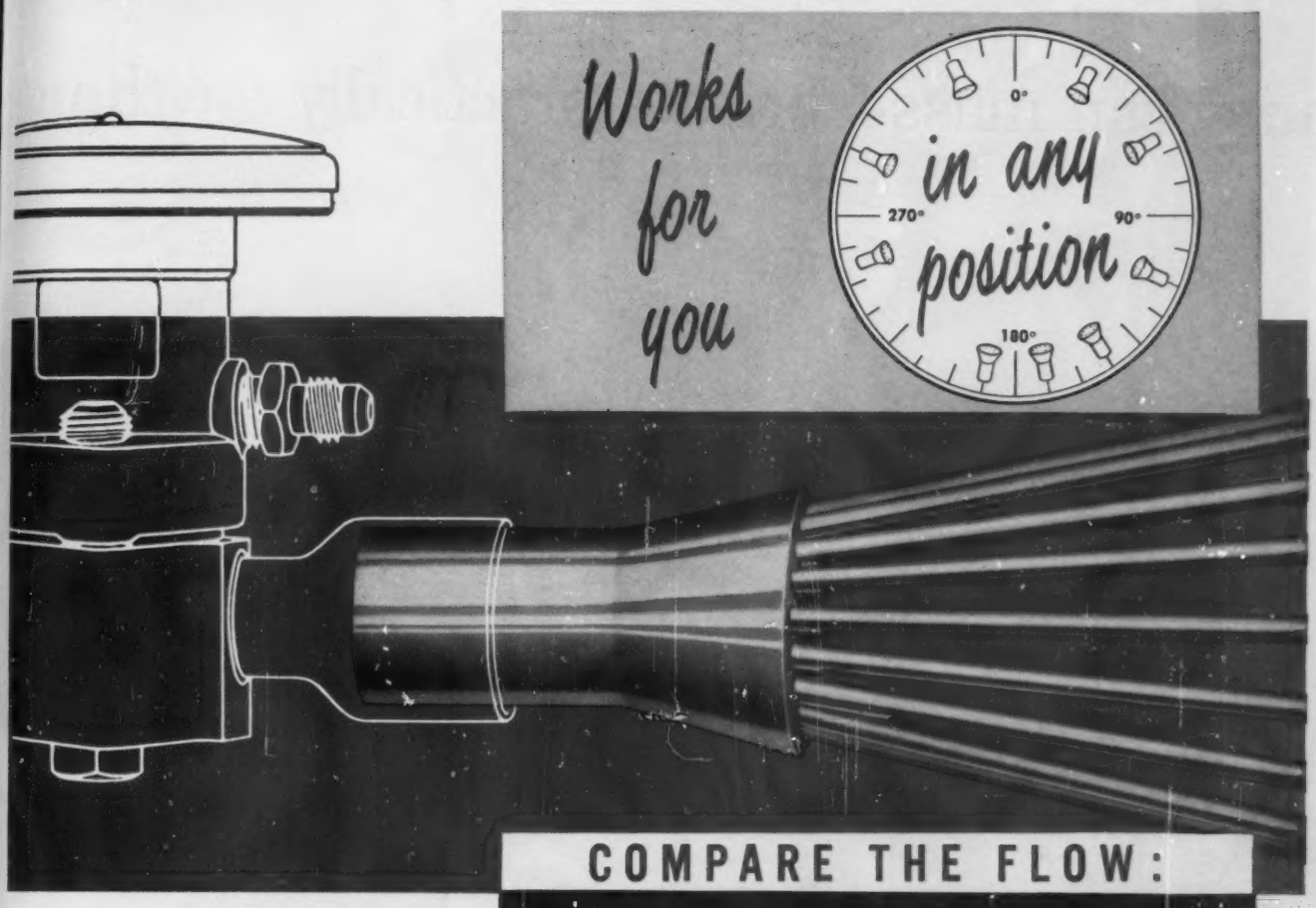
CHECK THE ADVANTAGES...THEN SPECIFY • INSTALL

ALCO'S NEW VENTURI-FLO® DISTRIBUTOR



Assures equal distribution to all passes of a multi-circuit evaporator with a low pressure drop.

WIDE APPLICATION RANGE—from 25% to 150% of Rated Capacity.

LOW PRESSURE DROP—permits closer, more economical Thermo® Valve sizing—assures more stable valve control.

ONE PIECE CONSTRUCTION—nothing to come apart, nothing to take apart—*No orifices, plates or nozzles needed* for capacity regulation.

COMPARE THE FLOW:

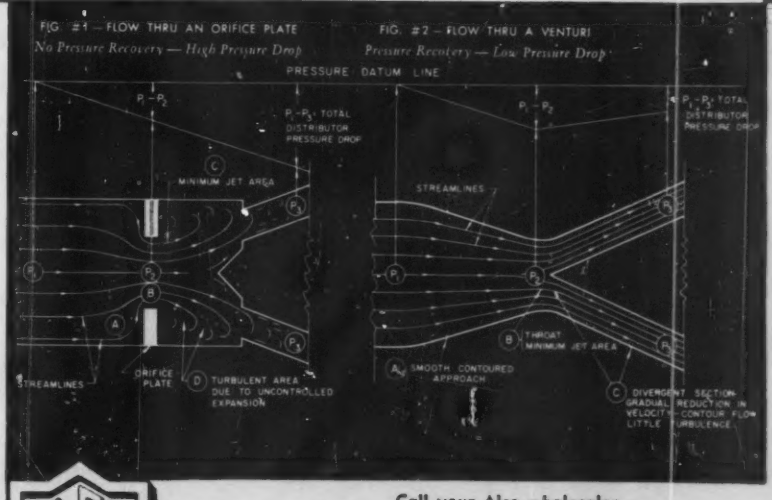


FIG. #1 — FLOW THRU AN ORIFICE PLATE
No Pressure Recovery — High Pressure Drop
FIG. #2 — FLOW THRU A VENTURI
Pressure Recovery — Low Pressure Drop
PRESSURE DATUM LINE
P1 – P2
P1 – P3 TOTAL DISTRIBUTOR PRESSURE DROP
MINIMUM JET AREA
STREAMLINES
ORIFICE PLATE
D TURBULENT AREA DUE TO UNCONTROLLED EXPANSION
B THROAT MINIMUM JET AREA
A SMOOTH CONTOURED APPROACH
C DIVERGENT SECTION GRADUAL REDUCTION IN VELOCITY—CONTOUR FLOW LITTLE TURBULENCE

Call your Alco wholesaler—
write for Specifications Bulletin No. 188.57.

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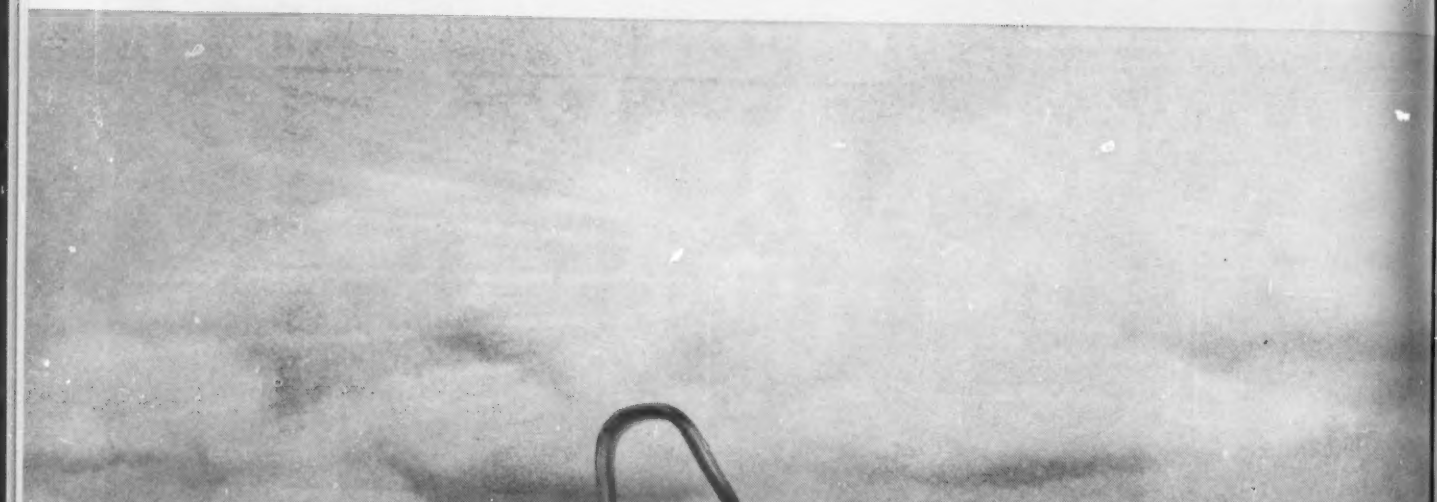
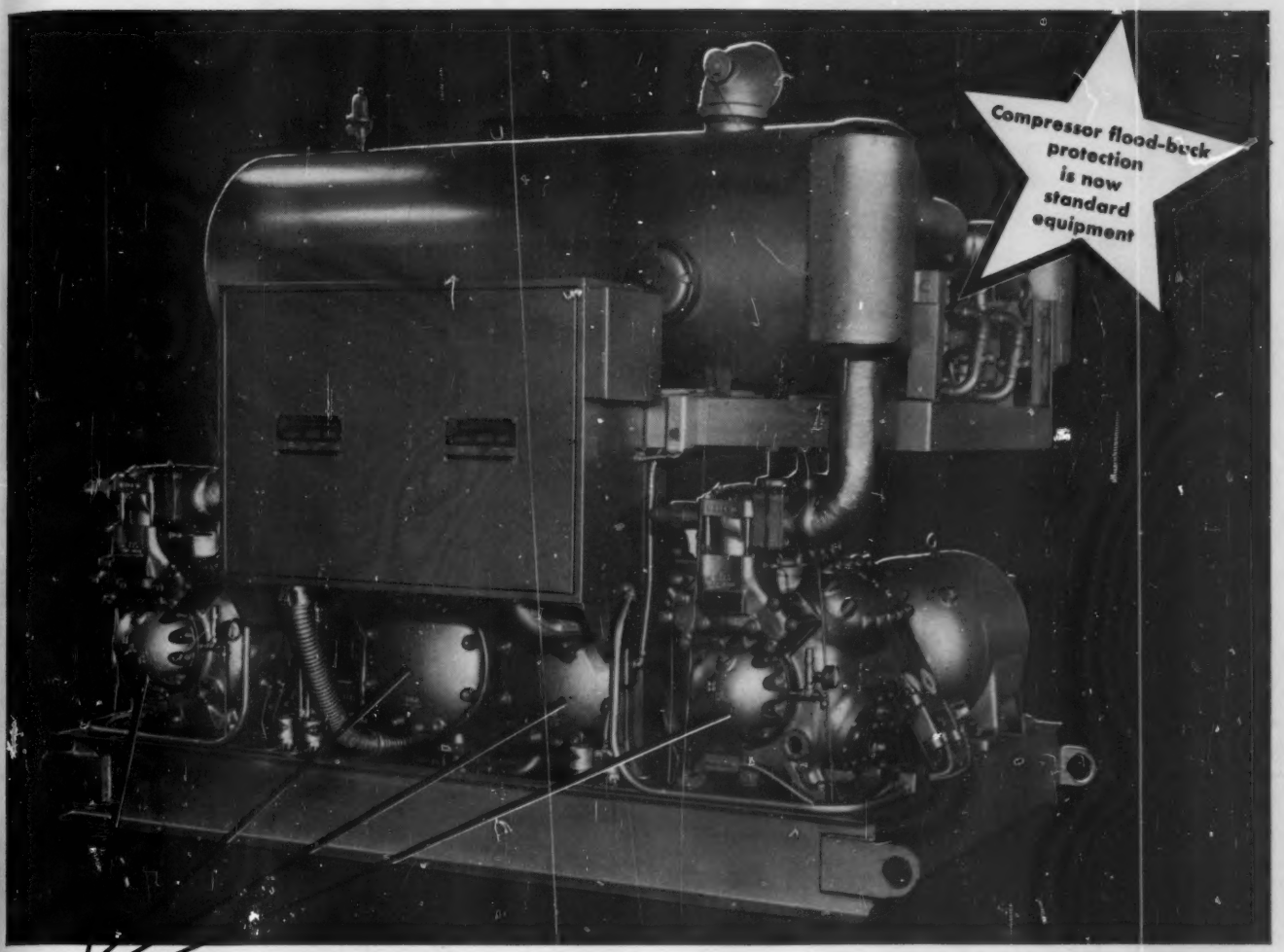
The one complete line of refrigerant controls: Thermostatic Expansion Valves • Refrigerant Distributors • Solenoid Valves • Suction Line Regulators • Flooded Evaporator Controls and Reversing Valves

Bundy can mass-fabricate practically anything





2 COMPLETE CIRCUITS WITH AUTOMATIC LEAD-LAG

The B&G Package Liquid Cooler is now available *with two complete refrigeration circuits*...dual compressors, dual condensers and a single evaporator! It has a capacity reduction of 25%—in stages of 100, 75, 50 and 25%. All operative and safety controls are installed and pre-wired.

An outstanding feature is the automatic lead-lag selector switch. In operation, No. 1 motor and compressor handle the load until cooling requirements are satisfied or until No. 2 unit needs to be called into use.

On the next starting, No. 2 unit picks up the load. *Therefore each motor and compressor gets equal usage!* This dual arrangement is also valuable because it provides stand-by protection in case of emergencies.

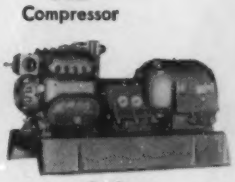
Remember, too, that all major parts of B&G "package" units are manufactured and guaranteed by Bell & Gossett Company...*one* responsibility for satisfactory operation.

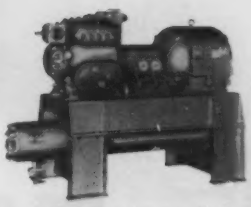
Send for complete engineering data on B&G Air Conditioning and Refrigeration equipment.

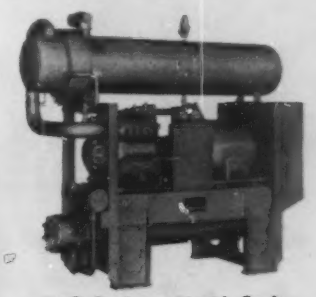
M-PAK
MODULAR PACKAGES

M-Pak units make it possible to build up from factory stock a comprehensive line of motor compressors, condensing units and package liquid coolers. Starting with five basic compressors, other stocked components are added to make up the unit desired. M-Pak Modular Packages are available for prompt delivery in capacities from 7½ to 150 tons.

B&G Compressor


B&G Motor Compressor

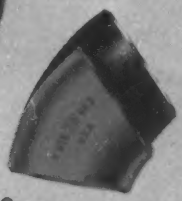

B&G Condensing Unit


B&G Package Liquid Cooler

BELL & GOSSETT
COMPANY
Dept. GX-64, Morton Grove, Ill.
Canadian Licensee: S. A. Armstrong, Ltd., 1400 O'Connor Drive, Toronto 16, Ontario

Now

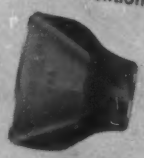
Compact 90° Elbow versus conventional long radius elbow*



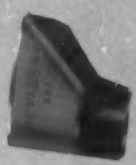
Compact 45° Elbow versus conventional 45° elbow†

Compact Straight Tee versus conventional

Compact Reducing Tee versus conventional



Compact Concentric Reducer versus conventional



Compact Eccentric Reducer versus conventional

Compact Welding Cap versus conventional

Note these actual photos showing dimensional differences between Compact Fittings and corresponding conventional fittings.

*The Compact 90° elbow is contrasted with a conventional long radius elbow which is by far the most commonly used type. Of course the dimensions of the Compact 90° elbow are the same as those of regular short radius Taylor Forge WeldELLS.

†The 45° conventional elbow has heretofore been obtainable only in the long radius pattern shown in the comparison.

Tests prove rugged strength

Photograph shows a test that has proved the rugged strength of Compact Line Fittings. The test unit was fabricated from 4" and 6" Standard Thickness (Schedule 40) Grade B pipe and includes a 6" x 6" x 6" Tee, a 6" 90° Elbow, a 6" 45° Elbow and a 6" x 4" Concentric Reducer.

In testing, a mean pressure of 875 psig (five times WOG rating) was established as basic pressure. Cyclic pressure variation was from 50% under to 50% over this pressure—in other words, from 437 psig to 1312 psig. The assembly was then subjected to 36,500 of these pressure cycles—the equivalent of one cycle per day for *100 years!*

Upon completion of the cyclic testing, the unit was hydrostatically burst, the break occurring, as it theoretically should, in the crotch of the tee and extending into the adjacent run and branch pipes. Moreover, the unit had to be brought to a pressure of 5200 psig before bursting, yet the theoretical bursting pressure of 6" pipe is only 5080 psig. After 36,500 pressure cycles this result is rather startling.

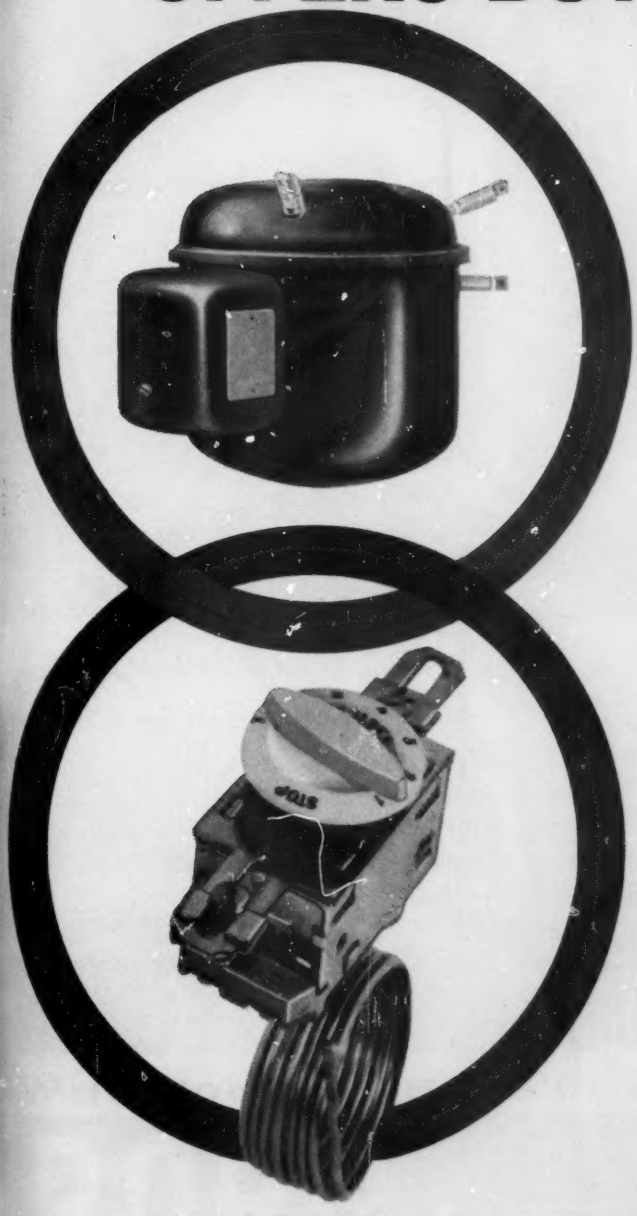
90° ELBOWS

45° ELBOWS

STRAIGHT TEES

it's the New TAYLOR FORGE

ONLY DANFOSS OFFERS BOTH...



compressor and thermostat, the heart and brain of your refrigeration systems. The Danfoss Pee-Wee Compressor and 51B Thermostat are just two products of a broad line of superior quality controls and equipment from this new source of supply. Together or individually, they offer special advantages in efficiency and economy important to all manufacturers.

Danfoss PEE-WEE Compressor: Compact and light (18 pounds), the Pee-Wee has demonstrated its superior quality and efficiency in over 3 million units in service around the world, under a variety of operating conditions. Available in 1/6, 1/8, 1/10 and 1/12 hp for low back pressure use and 1/6 and 1/8 hp for high back pressure use. Two-pole, high speed motor. Requires no oil or fan cooling. "Scotch yoke" construction provides efficient power transmission. Meets U. L. standards.

Model 51B Thermostat: An evaporator thermostat with full bellows construction for precise control of switching, the 51B Danfoss unit is a recent modernization of its famous Model 50 of which more than 8 million have been sold throughout the world.

Features of the 51B include: large cross-section (1 in. diameter) bellows with four pleat construction; rust proof housing of heavy gauge nickel plated brass; low height, matching requirements of modern insulating materials; one million cycle contact system in melamine insulation with single, double-break switch and silver plated contacts. (Contacts also cut in without rebound and, for instantaneous cut-out, a contact spring maintains full contact pressure until the circuit is broken.) U. L. listed.

Backed by a tradition and history of fine craftsmanship, Danfoss Inc. offers you the finest possible in refrigerating controls and equipment components.

Write for complete engineering information.

Danfoss INC.

115 Dell Glen Ave. • Lodi, New Jersey

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1230 Lake Shore Road E. • Port Credit, Ont.

11010

A SKILLED HAND IN CHEMISTRY . . . AT WORK FOR YOU

Now . . . Dimensional Stability at Subzero Temperatures

NOPCOFOAM™ H-602

(pour-in-place urethane foam)

This photo shows what happens when *(left)* standard fluorocarbon urethane and *(right)* Nopcofoam H-602 are subjected to −20°F for 1 hour. Cell walls of the standard urethane have collapsed—its distortions destroy the insulative value. But Nopcofoam H-602 maintains dimensional stability, allows no temperature leaks.

A further important advantage is low density. Until now it has been impossible to produce fluorocarbon-blown panels that would not shrink at low temperatures unless the overall density was 2.5 lb./cu. ft. In Nopcofoam H-602 we have succeeded in lowering the required in-place density by as much as 0.5 lb./cu. ft.—20% less weight and cost.

Nopcofoam is today's best temperature barrier

K-FACTOR COMPARISON

Nopcofoam H-602	.13
Polystyrene foam	.25
Fibrous glass	.25
Fibrous rock	.27
Cork board	.27

Btu/hr/sq ft/°F/in. at 75°F mean temperature

Write for complete technical information and call upon the experience of our Technical Service engineers.

NOPCOFOAM™ CHEMICALS
Pour-in-Place Urethanes

Plastics Division

NOPCO CHEMICAL COMPANY

North Arlington, N.J. • Chattanooga, Tenn. • Los Angeles, Calif.

SO VERY, VERY QUIET

BOHN-AIRE ROOM CONDITIONERS

"Whisper-quiet" and personalized control of room comfort are outstanding features of BOHN-AIRE Room Conditioners.

Mr. George Wenthe, one of the owners of the Effingham (Ill.) Holiday Inn Motel, says: "We had Bohn-Aire Room Conditioners installed in every guest room of our new motel. Bohn-Aire units are whisper-quiet and completely efficient; we are happy to endorse them for installations in which *exceptionally quiet operation* and *individual room control* are desirable features."

Extra slim! Extra compact! For concealed mounting (as above) or for exposed mounting in attractive beige enameled cabinets. In capacities from 200 to 1200 C.F.M.; ideally suited to apartments, motels, hotels, offices, hospitals, and similar structures. Available in 7 sizes for vertical mounting and 5 sizes for horizontal mounting, all with 3-speed push-button control.

SO QUIET
Bohn-Aire
SO EFFICIENT

Write for BOHN Bulletin 250 containing full details

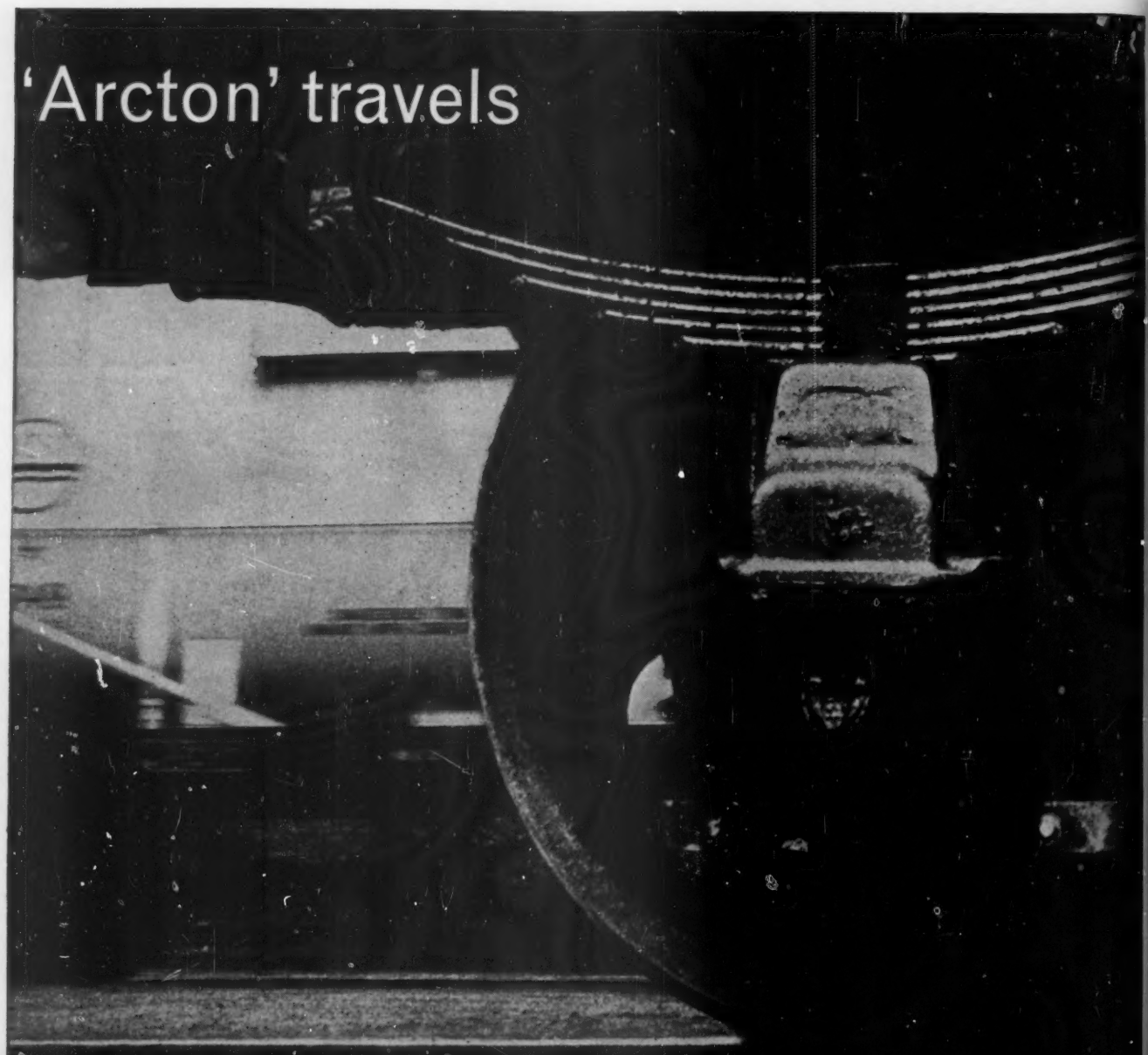
BOHN

Buy the known line... the BOHN line

ALUMINUM & BRASS CORPORATION

Danville Division • Danville, Illinois

'Arcton' travels



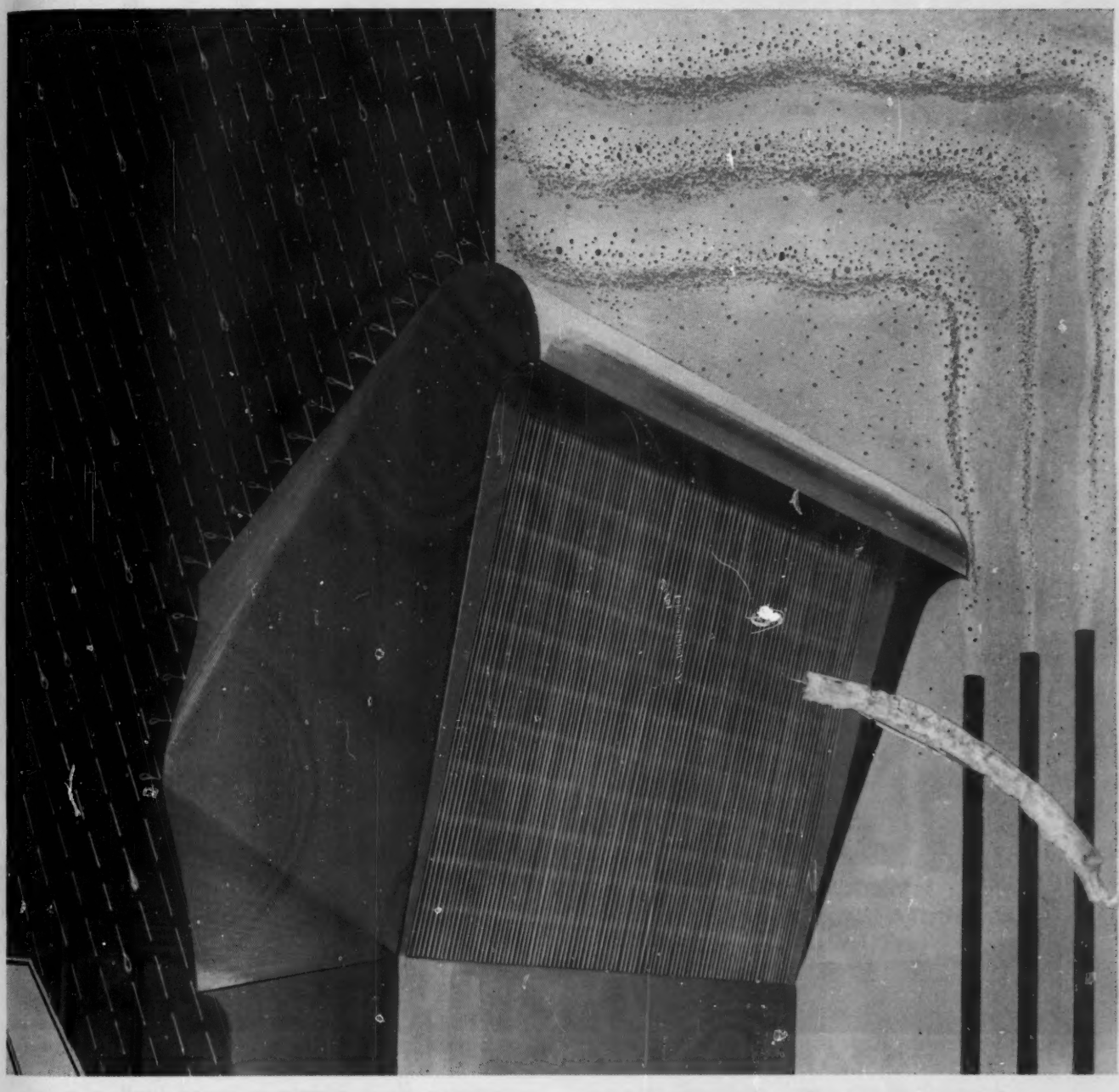
'Arcton' goes everywhere. It's not only Britain's best-selling refrigerant, but the choice of discerning users in Europe, India, Pakistan, South Africa, South America and as far afield as Hong Kong and Australia.
The reason is simple: for uniform high quality and constant performance the 'Arcton' range is unbeaten. There's an 'Arcton' chlorofluoro-hydrocarbon refrigerant for every use.
Literature and information are available on request.

ICI GENERAL CHEMICALS DIVISION

Imperial Chemical Industries Ltd., London, England.

AR 60

DESIGN HOUSINGS OF ANACONDA ALUMINUM FOR... EASIER, FASTER FABRICATION AND LASTING EYE APPEAL



Both you and your customers profit when you fabricate air-cooled condenser housings of Anaconda Aluminum. Your savings start with easier, faster forming and fabrication. You can give customers a piece of equipment which will enhance the appearance of their property. They save on upkeep since maintenance painting is not needed even in industrial atmospheres.

Installations can often cost less, too, when you take advantage of aluminum's light weight. Roof loadings are lower. Anaconda Aluminum can also show you how to get competitive advantages with other components of aluminum—end plates, fans, fin stock.

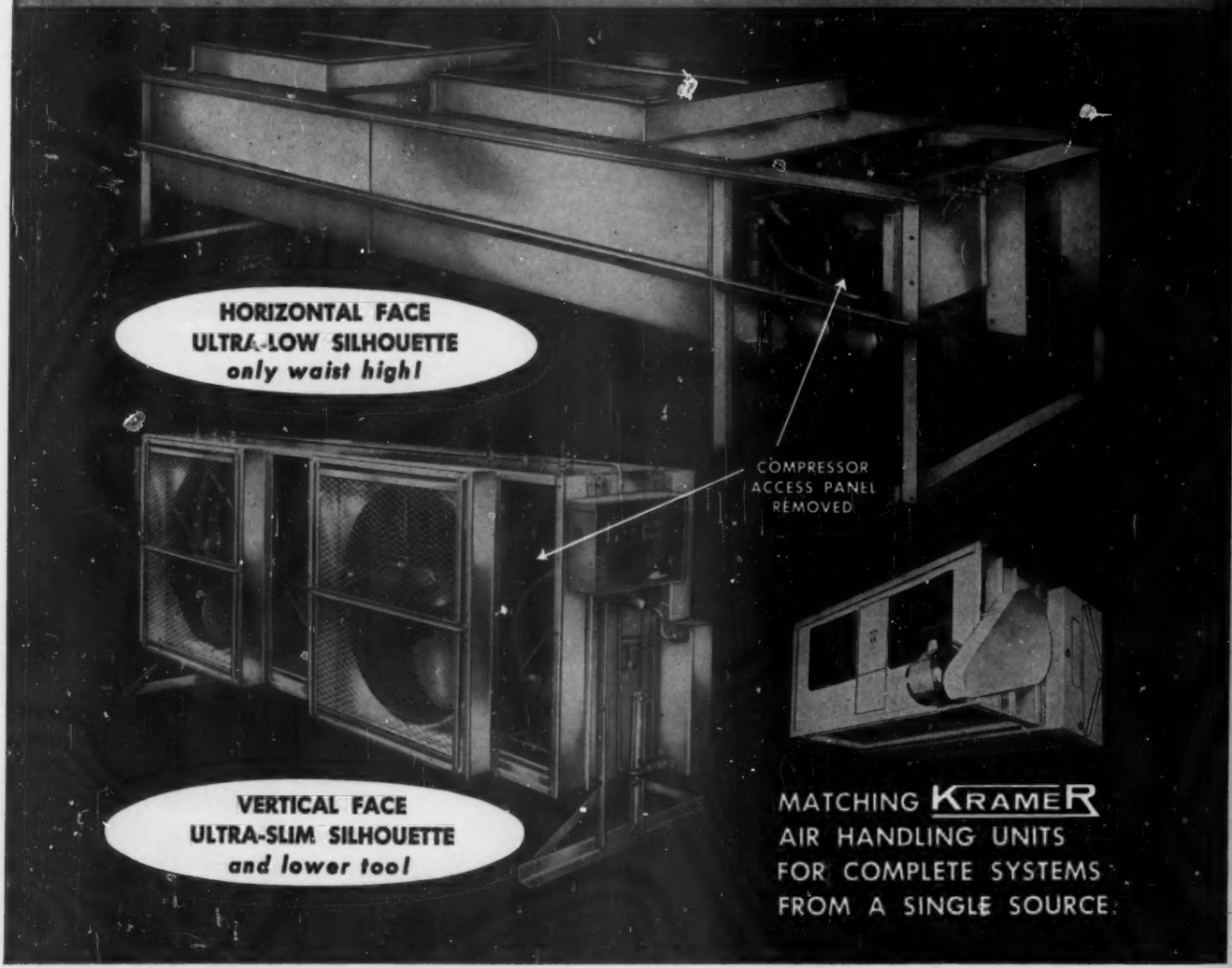
The exacting metallurgical control methods used by Anaconda Aluminum for fin stock have helped manufacturers of coils avoid many manufacturing problems. These same quality control methods, for all our sheet products, combined with customer-oriented service, may mean considerable savings in your manufacturing costs . . . equally important improvement in your products. Let us show you how! Call your nearby Anaconda Aluminum office or write to Anaconda Aluminum Company, Dept. AJ-12, Louisville 1, Kentucky.

ANACONDA ALUMINUM

From the leader in air-cooled condensing

the new KRAMER UNICON-COMPRESSOR

PACKAGED *WATER-FREE* CONDENSING UNIT



3 to 70 TONS—HORIZONTAL or VERTICAL FACE

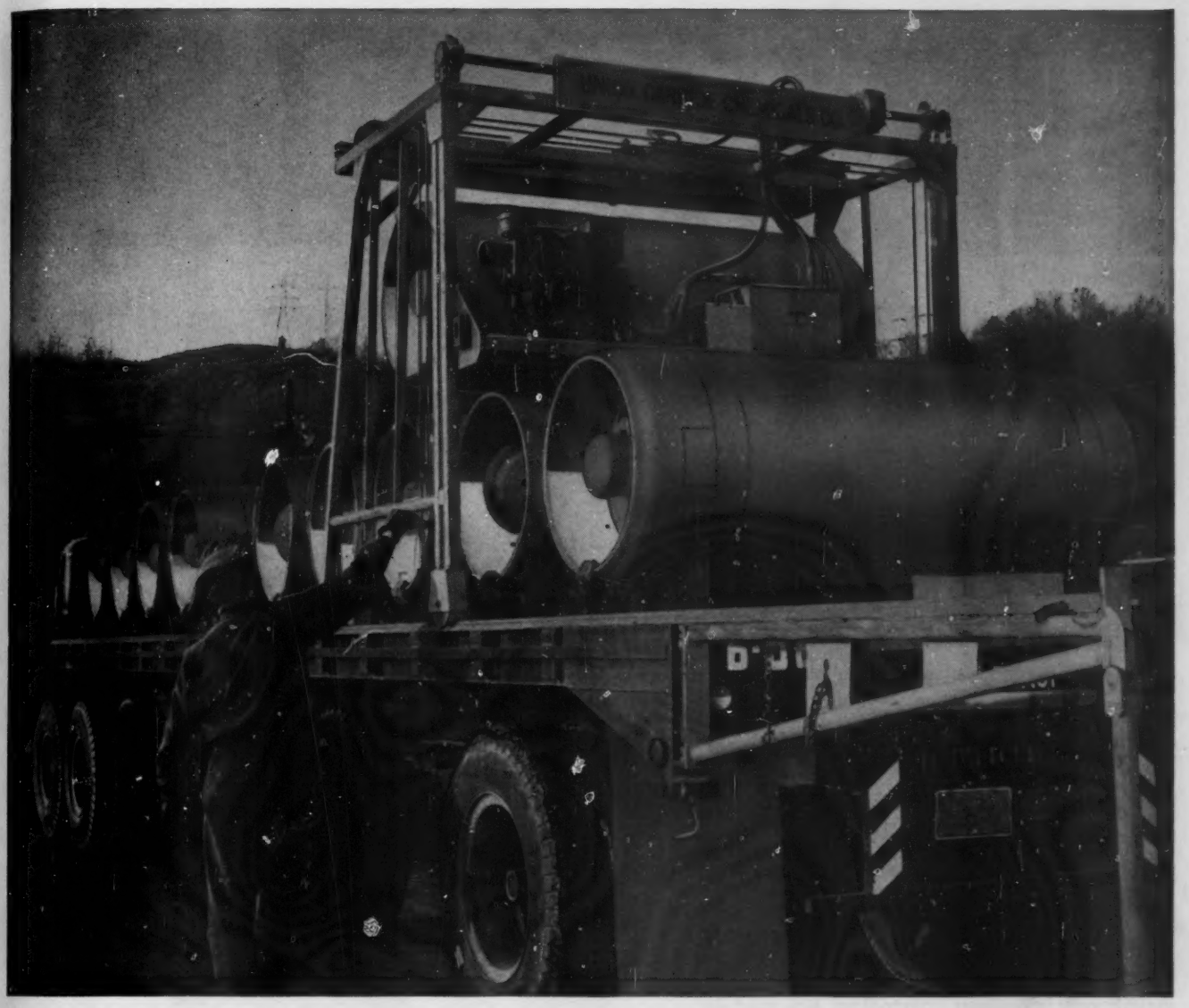
The low, slim-trim lines of the new Kramer air-cooled UNICON-COMPRESSOR give it unequaled flexibility in any architectural setting. Space saving outdoor design and low, low operating weight combine to reduce structural and engineering problems. Easy and economical to install, the complete packaged UNICON-COMPRESSOR has matchless accessibility for servicing. Its corrosion resistant aluminum casing, and its frame galvanized after fabrication, eliminate painting maintenance.

And remember—no place in the U. S. A. is too hot (or too cold) for the UNICON-COMPRESSOR. With the patented Kramer Winterstat it will operate any time of the year, without adjustment, even in the dead of winter.

Write for bulletin C460B.

KRAMER TRENTON CO., Trenton 5, N. J.

47 YEARS OF CONTINUOUS ACHIEVEMENT IN HEAT TRANSFER



The Fast, Convenient, Low Cost Way... Ton Cylinders Delivered by Ucon® RACK TRUCKS

refrigerants

Convenience is the word for the UCON Refrigerants Rack Truck...and *economy*, too. For refrigerant users preferring less-than-carload quantities, *and* individually packaged units, the Rack Truck reduces in-plant handling problems, eliminates need for special bulk storage facilities.

One-ton cylinders are unloaded directly onto fork-lift trucks or other haul away devices with the Rack Truck's own hoist mechanism...or if space permits, the Rack Truck can be driven right into your warehouse storage area.

Result: substantial savings in freight costs, greater handling safety, reduced outlay for cylinder deposit, and far fewer man-hours required for unloading, testing and handling...more profits for you!

Your local UCON Refrigerants Representative has the whole, eye-opening story on the big savings that can be yours with convenient Rack Truck deliveries. Call him today...and put this money-savir*g* fleet to work for *you!*

UNION CARBIDE

Ucon® refrigerants

These 5 UCON Refrigerants will meet your refrigeration and air conditioning needs

- UCON Refrigerant 11 Trichloromonofluoromethane
- UCON Refrigerant 12 Dichlorodifluoromethane
- UCON Refrigerant 22 Monochlorodifluoromethane
- UCON Refrigerant 113 Trichlorotrifluoroethane
- UCON Refrigerant 114 Dichlorotetrafluoroethane

UNION CARBIDE CHEMICALS COMPANY, *Division of Union Carbide Corporation,* 270 Park Avenue, New York 17, New York

FIRST

completely self-contained air conditioning units

GOVERNAIR CORPORATION

. . . began the manufacture of the first completely self-contained commercial type air conditioning units in 1937. Development has been continuous since that time, and Governair now manufactures packaged air conditioning units, for both commercial and industrial application, in the widest range of capacities available in the industry.

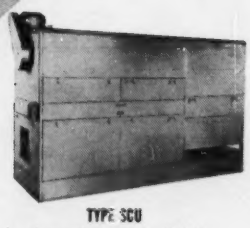
Governair Corporation also constantly recognizes the importance of producing a high quality, completely flexible system to benefit the end user best. Production facilities are tailored to accomplish these ends economically. This, in turn, makes the Governair prices highly competitive, even though only the finest components are used in the assembly.

Development of the larger tonnage units has made available the economy of installation and reliability of a factory balanced and tested package system . . . for even the largest industrial requirements. Single units of 120 ton capacity are being fabricated regularly — and to the exact specifications of the customer.

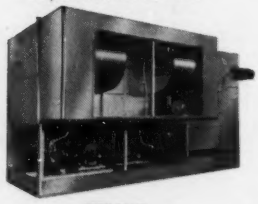
GOVERNAIR CORPORATION
4840 NORTH SEWELL
OKLAHOMA CITY, OKLAHOMA

GOVERNAIR

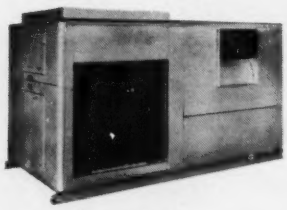
TO MANUFACTURE TO DEVELOP



TYPE SCU
Evaporative Condenser Model

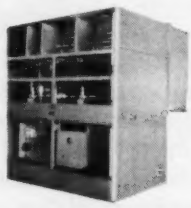


TYPE SCMZ
Complete Multi-Zone Package



TYPE SCAC
Air Cooled Condenser

TYPE ST
Water Cooled Model



TYPE STMZ
Water Cooled Multi-Zone

Write for detailed information, or consult your yellow page directory for the name of our local representative.

FIXED SETTING CONTROLS

New Ranco D60/D62

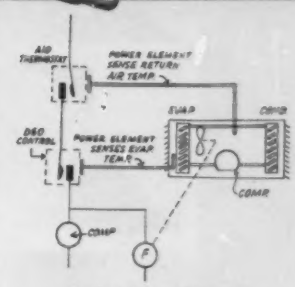
RANCO D60 CONTROL
Switch action is Single Pole, Single Throw. Circuit opens on decrease in temperature.

ideal for refrigeration and air conditioning applications requiring **Wide Differential: 20°F to 45°F**

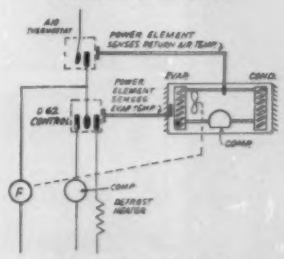
New Ranco D60/D62 controls are specially designed for refrigeration and air conditioning applications—wherever relatively wide differential and fixed setting (no range) are required. *Differentials* available (not adjustable) are 20°F (Minimum) to 45°F (Maximum). Other differentials, above or below listed temperatures; may be provided. Operating temperature limits are −15°F (Lowest cut-out) and +80°F (Highest cut-in), not in the same control. Factory tolerance is ±1°F.

D60/D62 controls are actuated directly by vapor-pressure filled bellows, have no dial shaft or "off" position. *Increase in temperature* at the power element capillary tube bulb is transferred directly to toggle mechanism. *Decrease in temperature* permits bellows to be depressed to toggle mechanism in reverse direction.

For additional details, contact Ranco Incorporated, Columbus 1, Ohio.



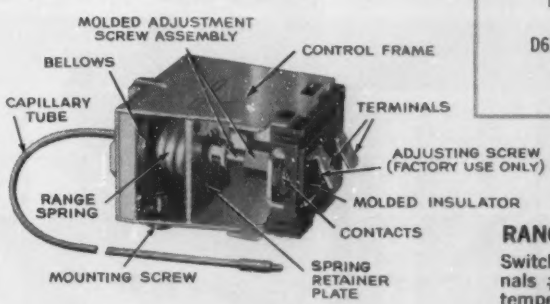
Typical Wiring Diagram
D60 EVAPORATOR DEFROST CONTROL
Without Defrost Heater



Typical Wiring Diagram
D62 EVAPORATOR DEFROST CONTROL
With Defrost Heater

RANCO D60/D62 CONTROLS are highly recommended for these product applications:

- Air Conditioning Evaporators
- Evaporator Defrost Control Without Defrost Heater
- Evaporator Defrost Control With Defrost Heater
- Suction Line Temp. Control
- Control of Auxiliary Heaters In Heat Pumps
- Heat Boost Control During De-Icing Cycle In Heat Pumps



ELECTRICAL RATINGS

	Volts, a.c.	Maximum Motor AMPERE Rating Full Load	Locked Rotor
D60 Control			
Inductive	120/240	16	80
Resistive	250	16	—
Pilot Duty	240*	—	—
D62 Control (Terminals 2 & 3)			
Inductive	120/240	20	78
Resistive	250	20	—
Pilot Duty	240*	—	—
D62 Control (Terminals 1 & 2)			
Resistive	250	20	—
Pilot Duty	240*	—	—

**Volt amperes @ 240V, a.c.*

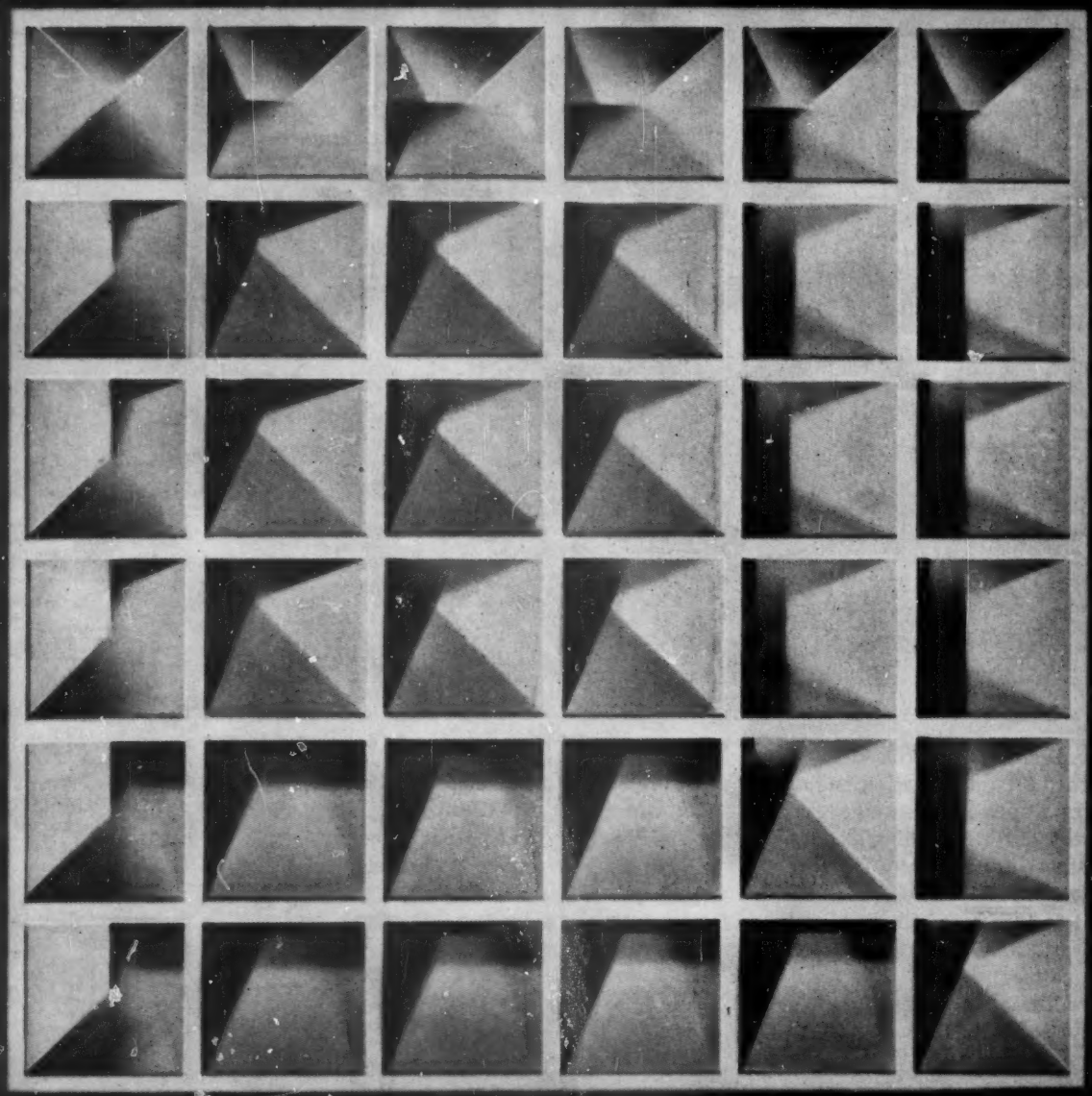
RANCO D62 CONTROL

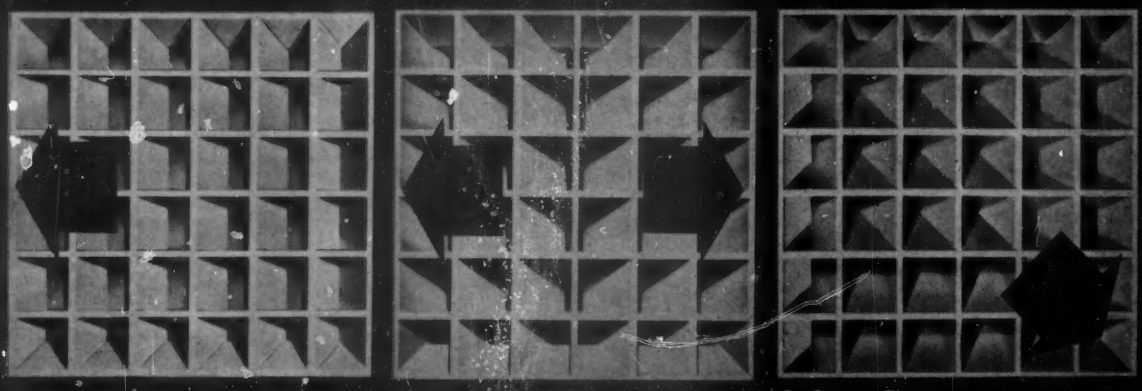
Switch action is SPDT. Circuit to terminals #1 & #2 **closes on decrease** in temperature (opens on increase). Circuit to terminals #2 & #3 **closes on increase** (opens on decrease). Either #1 or #3 terminal may be removed for SPST action.

Ranco® INCORPORATED
Columbus 1, Ohio

MORE THAN 100 MILLION RANCO CONTROLS NOW IN USE

"I am a Carnes Diffuser Module" (corner throw, actual size)





A: 1-way throw

B: 2-way throw

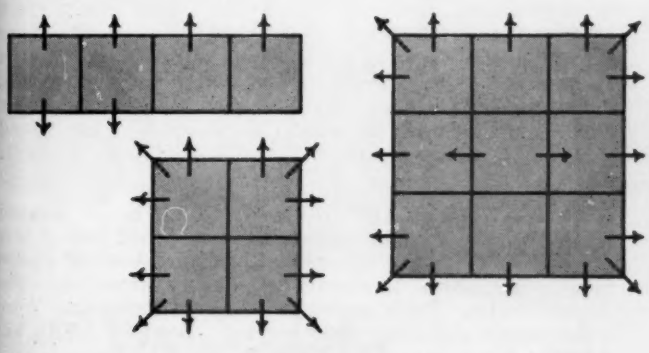
C: Corner Throw

"By combining multiples of my three basic air throw patterns, I can solve almost any air handling problem"

only a CARNES modular diffuser can make these statements

I am unobtrusive My small texture makes me blend inconspicuously with any ceiling material. I fit flush with the ceiling, too, and four of me make a square 12" x 12" — the size of a standard ceiling tile. I have very little see-through.

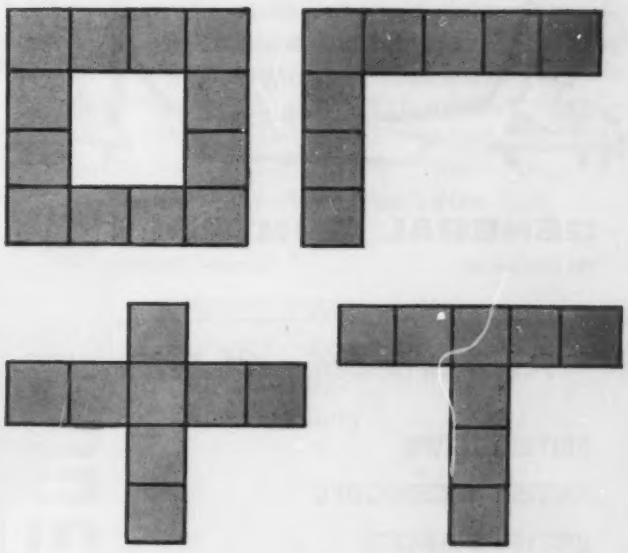
I am versatile By combining any one of my 3 air patterns, shown on the opposite page, I form an infinite number of sizes and shapes, to direct air in 1, 2, 3 or 4 directions for m ximum efficiency.



I fit any design My square shape and size (6" x 6") make me an ideal module for any architectural design. Just check the ways I can be used: butted to lights, spacers between lights, alternating line with lights, continuous strips around ceiling coves, along corner windows, around columns.

I am interchangeable Just snap me into my simple frame. I may be easily removed to change direction of air throw made necessary due to rearrangement of occupied areas, or for periodic cleaning.

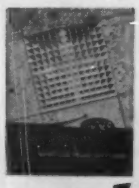
I make any shape That is, except a circle.



I work with any ceiling Plaster, tile, metal pan, "T" bar, spline — you name it and I have a frame that works efficiently with it. I have a frame to fit any combination of shapes or sizes you want to use me in.

I am tough Carnes makes me out of General Electric's LEXAN® — that's a high impact, fire retardant material with excellent color and dimensional stability. I'll give years and years of service. ®Registered Trade Mark

Get more of my intimate details by writing for catalog no. 460



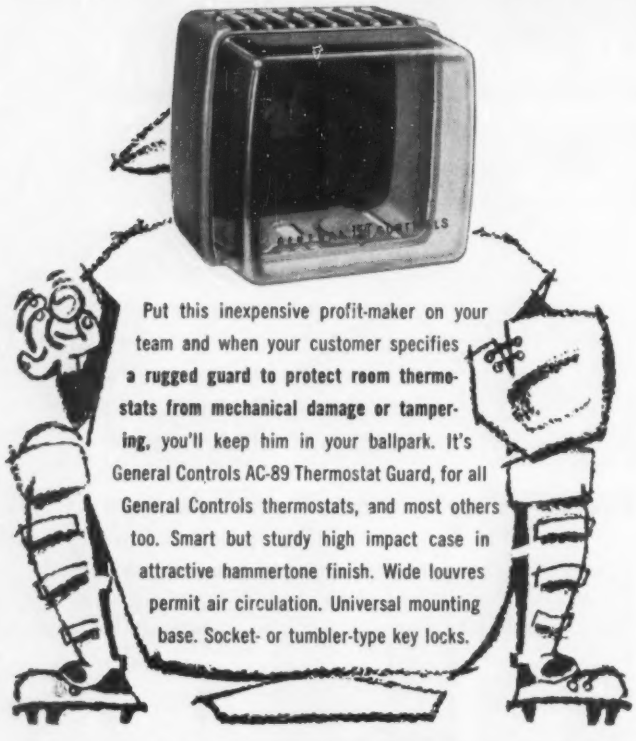
Carnes also manufactures: Round, Rectangular, Square, Convertible and Baseboard Diffusers • High-Velocity ATC Units • Low-Velocity ATC Units • Registers • Grilles • Roof and Wall Ventilators

CARNES CORPORATION, VERONA, WIS

CARNES

Canadian Subsidiary: Weir Air Equipment Company, Ltd. Winnipeg, Canada.

GUARD THAT CUSTOMER!



Put this inexpensive profit-maker on your team and when your customer specifies **a rugged guard to protect room thermostats from mechanical damage or tampering**, you'll keep him in your ballpark. It's General Controls AC-89 Thermostat Guard, for all General Controls thermostats, and most others too. Smart but sturdy high impact case in attractive hammertone finish. Wide louvres permit air circulation. Universal mounting base. Socket- or tumbler-type key locks.

Call your local factory branch office, or write :

GENERAL CONTROLS

801 Allen Avenue Glendale 1, California

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NO DISPLAY advertising at this heading.

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PRODUCT DEVELOPMENT ENGINEER — wanted must have experience in filtration to develop high efficiency filter media and equipment, new types of filters, and new applications for existing products. Market orientation desirable. Salary will be commensurate with background and experience. Resume to: Mr. David A. Greason, Owens-Corning Fiberglas Corp., Newark, Ohio

SALES ENGINEER — Challenging opportunity for growth with leading manufacturer of heat exchangers, pumps, and hot water heating equipment for a man experienced in selling specification equipment to wholesalers, contractors, and engineers. Engineering or technical degree or equivalent experience. Willing to relocate and travel. Send resume to Box 202 ASHRAE JOURNAL.

PRODUCT ENGINEER — national company seeking man with experience in application and/or product design engineering for air conditioning, refrigeration controls, valves and/or fractional HP hermetic compressors. New position with growth potential. Northeast location. Send resume and salary desired. Box 229 ASHRAE JOURNAL.

MANUFACTURERS' REPRESENTATIVE — wanted by established national manufacturer of broad line of low pressure steel heating boilers and packaged heating equipment. Exclusive territories available throughout the United States. Reply giving information concerning your organization and other lines carried. Box 231 ASHRAE JOURNAL.

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REFRIGERATION AND AIR CONDITIONING ENGINEER, BSME, — with MA degree in General Business looking for a challenging position in technical management — product development, planning or sales. Excellent education record and 5 years of R&D experience in refrigeration and air conditioning. Resume available. Box 232 ASHRAE JOURNAL.

MANAGING ENGINEER — with well balanced design, manufacturing and application experience. Can assure

NEW GM-DELCO THRIFT-PAK

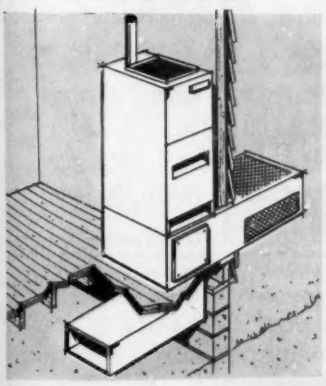
OPENS THE MARKET FOR LOW-COST HOME CONDITIONING

GM-Delco's new thru-the-wall heating and cooling system now brings the vast $10,000 to $15,000 range of homes into the air conditioning market. In fact, *Thrift-Pak* was specifically designed to make it practical to install year-round air conditioning in these modestly priced homes. Low installation cost makes this possible. In just sixty minutes, two men can install this new GM-Delco Conditionair. For more information, write the Delco Appliance Division, Dept. J-7, General Motors Corporation, Rochester 1, New York.

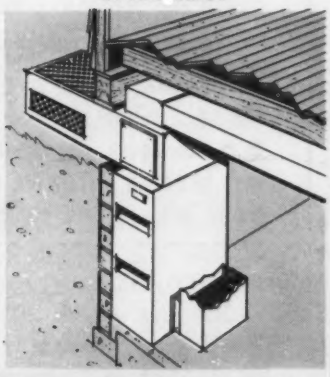
THRIFT-PAK
(upflow or counterflow models)
gives complete heating and
air conditioning

Versatile - 4 practical applications

CRAWL SPACE



BASEMENT



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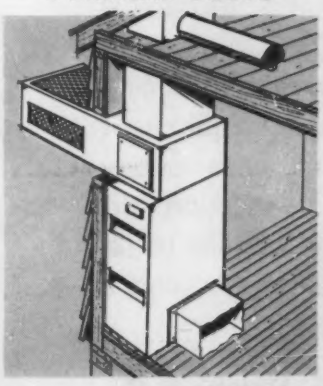

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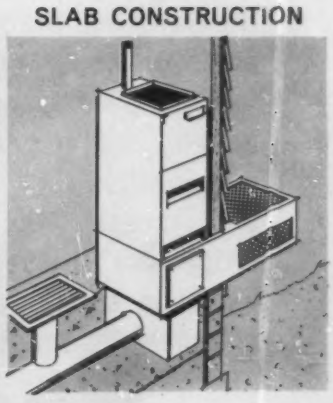

GM Delco 365 CONDITIONAIR

DELCO APPLIANCE DIVISION • GENERAL MOTORS CORPORATION • ROCHESTER, NEW YORK

NEW! Waterproof!

NOW STANDARD ON ALL REGULARLY CATALOGUED JE SOLENOID VALVES

New epoxy-clad coils are line-tested at 2300 volts completely submerged in water. Eliminate coil failure! Impervious to almost anything and everything—condensation, fluid spray, frost, oil, gasoline and other corrosive liquids and vapors. Unlimited indoor or outdoor applications. U.L. listed. Wire, write, or phone:

CONTROLS DIVISION **JACKES EVANS MFG. CO.**
4427 GERALDINE AVENUE • ST. LOUIS 15, MO.

successful organization or operation of a division to manufacture Air Conditioning, Refrigerated and related products. Experience in field broad and extensive. Preference, but not sole interest, is engineering and application of commercial and industrial equipment and operation of a top-notch refrigerating system processing facility. Box 233 ASHRAE JOURNAL.

FIELD SERVICE REPRESENTATIVE — experienced service engineer desires challenging position, 15 years experience all phases commercial refrigeration and air conditioning; Associate Member ASHRAE 11 yrs. Certificate Member of RSES 10 yrs. Presently employed R&D. Travel eastern half U. S. Available January 1, 1962. Box 235 ASHRAE JOURNAL.

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Our Circulation Department has recently mailed out Business Reply Cards. Verification of our Circulation is obtained by this means. Please take a minute of your time *now* to check off the information requested. Your cooperation in signing and returning these cards immediately will be greatly appreciated.

SALES ENGINEER

Wanted by well established San Francisco firm to call on mechanical engineers and contractors in the Bay Area. Graduate mechanical engineer under 40, with at least five years of experience in selling heating and ventilating equipment preferred. Excellent base salary and incentive compensation arrangement. If you would like to relocate in sunny California, send a complete resume to:

Norman S. Wright & Co.
1015 Folsom Street
San Francisco, California

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Copies of 1960 edition of ASHRAE TRANSACTIONS are still available. Following precedent volumes, this issue is handsomely bound in maroon cloth, size 6 by 9 inches. Included is the complete record of 1960 Society events . . . the Semiannual Meeting in Dallas, the Annual Meeting in Vancouver (including technical papers presented at both National Meetings and discussions upon them), listings of national officers and committees, Chapter personnel, By-Laws, Financial statement, report of Research and Technical Committee, and obituaries.

Previously, TRANSACTIONS was distributed to Society members, upon specific request, without charge. The 1960 edition is priced at $3.00 to members, $6.00 to non-members. The quantity is limited and those who wish to obtain a copy are requested to return the form shown below, at the earliest possible date.

Name .....

Address .....

I wish to have a copy of the ASHRAE 1960 TRANSACTIONS. You may bill me $3.00 ($6.00 for non-members).

Signature .....

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS
United Engineering Center,
345 E. 47th St., New York 17, N. Y.

...I can get just the fin spacing, tonnage and CFM I want."

Now, Halstead & Mitchell gives you a choice of 6, 7, 8 or 10 fins per inch . . . makes it much easier to match a central fan-coil unit to a particular job or to meet specifications.

And look what else H&M offers: a selection of three coil face areas for each model; direct expansion or chilled water cooling coils (1 to 8 rows); hot water, standard steam or non-freeze steam heating coils (1 or 2 rows); cooling capacities of 3 to 92 nominal tons with 880 to 47,750 CFM; Turbu-Flo coils arranged for right or left-hand connections; horizontal or vertical mounting for discharge in any direction . . . and every accessory item you need to simplify your installation.

Why don't you check H&M Central System Air Conditioners? Call your wholesaler or write for Bulletin AHU-100. Halstead & Mitchell Co., Dept. E-12, Bessemer Bldg., Pittsburgh 22, Pa.

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AIR HANDLER

HM
Halstead & Mitchell

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Central System Air Conditioners • Air-Cooled Condensers • Cooling Towers • Water-Cooled Condensers

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For Sub-Zero or High Temperatures

One of many Rempe Coil jobs: This Tank and Coil assembly was built for use with thermostatic expansion valves to dewax oil at minus 35° F. Any diameter or any length to your specifications or our design. Can be supplied in Steel, Alloy, Stainless Steel, Copper or Aluminum.

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For Residential, Commercial or Industrial Applications

This is the popular Rempe Hi-Cap line of heating coils. Equipped with the exclusive "Defrost-Tap" to keep coil warm during steam off-cycle and prevent freeze-up.
Single or double row designs available . . . any face area desired.
Eight, collared, Aluminum fins per inch on copper tube. Heavy steel casings.

Engineering Data Book

Pipe Coil and Fin Coil Designs. Heat Transfer Coefficients. Get a copy for your library.

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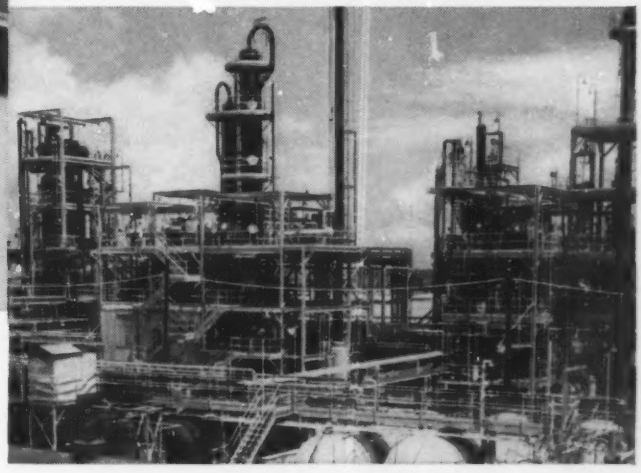

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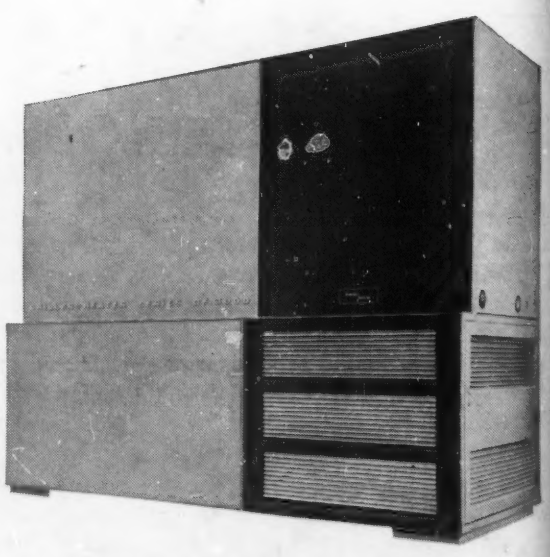

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